JOURNAL
OF THE
FRANKLIN INSTITUTE
OF THE STATE OF PENNSYLVANIA,
FOR THE PROMOTION OF THE MECHANIC ARTS.

VOL. CXX. OCTOBER, 1885. No. 4.

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ON TIDAL THEORY AND TIDAL PREDICTIONS.

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[*Continued from Volume CXIX., page 188.*]

II. TIDAL PREDICTIONS.

For the computations described below, there were available about 1,100 high water and as many low water observations of time and height, made at the Government pier near Cape Henlopen, and about 1,500 high water and as many low water observations of time made at various points of Delaware Bay and river as far up as Bordentown.

The first question to be decided in regard to the methodical treatment of these observations, was to which lunar transit they were to be referred; that is, which lunar transit was to be considered as having generated the high water. This question is not a new one; under the name of "the age of tides" it has been discussed, for the European coast, by such authorities as Laplace,

Lubbock and Whewell. Laplace sought to determine the generating lunar transit by means of the time at which the extremes of semi-menstrual inequality—Spring and Neap tide—make their appearance. Lubbock, in addition to these, drew within the scope of his investigation the inequalities caused by lunar parallax and declination and, on the basis of very extensive series of observations, attempted to discover the best fitting lunar transit. Whewell caused simulantaneous tidal observations to be made at a great many points of the European as well as the American coast of the Atlantic Ocean, and by the aid of these constructed his "cotidal lines," with a view thus to determine the place of origin of the Atlantic tidal wave.

None of these various investigations have rendered a definite result, and up to the present we have no certainty as to which lunar transit is the generating one. In the following computations, the immediately preceding transit has been chosen, not from the writer's conviction that it is the correct one, but simply because he did not find himself in a position to improve on a choice which has been made with very general consent for our Atlantic coast. This transit has been found to agree very well with the time of appearance of the inequalities caused by lunar declination (that is, the diurnal inequalities), while the extremes of the semi-menstrual inequalities (that is, Spring and Neap tide), as referred to it, occur about one day too late, thus pointing to an earlier lunar transit as the generating one.

As a preliminary step towards the methodical treatment of the above-mentioned observations, the so-called "first reduction" of them was made according to the following schedule for high water, a similar one being employed for low water:

Observations of High Water at Iron Pier, Cape Henlopen.

FIRST REDUCTION.

Observed Date & Time of High Water.			Date & Time of Generating Lunar Transit.			Lunitidal Interval.		LUNAR.		Solar Declination.	Observed Reading of Gauge at High Water.
								Parallax.	Declination		
June	h.	m.	June	h.	m.	hrs.	min.	minutes.	degrees.	degrees.	feet.
2	5	45	1	21	29	8	16	60½	+14	+22	6·50
2	18	20	2	9	58	8	22	60½	+16	+22	5·75

It should be noted here that an effort has been made to determine the influence of the Sun's distance, which, however, failed entirely, owing probably to the limited number of observations available; no note is therefore taken in the schedule of solar distance.

From the "first reduction," an epitome was made of the lunitidal intervals of high water and of low water (the so-called "second reduction"), grouping them in both cases according to the hour of the generating transit in twelve columns, the low water always being referred to that transit which generated the preceding high water. The grouping into twenty-four hour columns was attempted, but had to be abandoned, the number of observations not being sufficient to render reliable means in so many columns. The supposition, therefore, is that the inequalities of time from the thirteenth to the twenty-fourth hour, are an exact repetition of those from the first to the twelfth hour, with this easily-understood exception, that the influences caused by solar declination will appear reversed, in the same way and for the same reasons that the influences of lunar declination appear reversed in the two lunar groups. This supposition, although probably not entirely correct, was the nearest approximation to truth to be obtained under the circumstances. In the second reduction of high water lunitidal intervals thus made, lunar parallax and solar declination were noted for each lunitidal interval, they principally influencing the time of high water, and in the second reduction of low water lunitidal intervals thus made lunar parallax, lunar declination and solar declination were noted for each lunitidal interval, they principally influencing the time of low water. The observations were grouped so as to render apparent the amount of these various heavenly influences, and the following tables show the means thus obtained in the second reduction of high water lunitidal intervals :

The Influence of Lunar Parallax on the Time of High Water at Cape Henlopen.

MEANS OF SECOND REDUCTION.

Table 1.—The Higher Parallaxes.

Hour of Moon's Transit.	Number of Observations.	Luni-tidal Interval	Lunar Parallax.	Solar Declination.
		hrs. min.	minutes	degrees.
1	51	8 18	59·4	+ 0½
2	50	8 08¼	59·2	— 0½
3	45	8 01½	59·0	0
4	48	7 53½	58·8	— 1½
5	47	7 56	58·3	— 2⅓
6	50	8 01	58.3	— 2
7	50	8 14	58·2	— 1½
8	43	8 27	58·3	— 2½
9	46	8 37½	58·9	+ 0½
10	43	8 35	59·4	— 1
11	40	8 33½	59·4	— 2⅙
12	47	8 26½	59·3	+ 0¼

Table 2.—The Lower Parallaxes.

Hour of Moon's Transit	Number of Observations.	Luni-tidal Interval.	Lunar Parallax.	Solar Declination.
		hrs. min.	minutes.	degrees.
1	40	8 25	54·7	— 3
2	43	8 15½	54·8	+ 0¼
3	49	8 01	54·8	— 0½
4	58	7 55	55·0	— 1⅔
5	49	7 54¾	54·9	+ 0⅙
6	52	8 03	55·0	— 2
7	49	8 15¼	54·9	+ 0⅓
8	41	8 37½	54·8	— 2⅓
9	40	8 49½	54·8	— 1
10	43	8 49½	54·8	+ 0¼
11	35	8 47½	54·7	+ 0⅓
12	42	8 39¼	54·8	+ 2⅚

The Influence of Solar Declination on the Time of High Water at Cape Henlopen.

MEANS OF SECOND REDUCTION.

Table 3.—The Sun declines North.

Hour of Moon's transit.	Number of Observations.	Luni-tidal Interval.	Lunar Parallax.	Solar Declination.
		hrs. min.	minutes.	degrees.
1	43	8 21⅓	57·4	+14
2	48	8 08½	57·0	+13¾
3	45	7 57¼	57·0	+15¾
4	48	7 46¾	56·9	+15
5	46	7 52	56·5	+14½
6	46	7 54	56·6	+14¾
7	47	8 07½	56·5	+14½
8	38	8 29½	56·6	+13⅓
9	40	8 43	57·2	+15¾
10	43	8 44	57·0	+14
11	37	8 39½	56·9	+13⅔
12	50	8 33½	57·2	+13⅔

Table 4.—The Sun declines South.

Hour of Moon's transit.	Number of Observations.	Luni-tidal Interval.	Lunar Parallax.	Solar Declination.
		hrs. min.	minutes.	degrees.
1	48	8 21⅓	57·3	—14⅔
2	45	8 12⅔	57·3	—14½
3	49	8 04½	57·2	—15
4	58	8 00	56·9	—16
5	50	7 58	56·7	—15
6	56	8 08½	56·6	—16
7	52	8 20⅓	56·6	—14⅓
8	46	8 34	56·5	—15
9	46	8 43	56·8	—14½
10	43	8 40⅓	57·3	—15
11	38	8 41	57·4	—14¾
12	39	8 31⅓	57·2	—13⅔

In these tables, as in all following ones, the sign + prefixed to solar declination, indicates that it is *northerly*, and the sign — prefixed to solar declination, indicates that it is southerly.

It will be noticed that in Tables 1 and 2, the differences in the means of solar declination are only slight; the differences, then, appearing in the length of the lunitidal intervals of one and the same hour in the two tables, must be attributed almost entirely to the differences in lunar parallax; in a similar way, the differences in the length of the lunitidal intervals of one and the same hour in Tables 3 and 4 must be attributed to the differences in solar declination, lunar parallax being practically the same in both tables. By means of a judicious method of approximation, we arrive at the following results, in regard to the influence of the two above-mentioned heavenly conditions:

Table 5.—Differences in Time of High Water per Minute of Lunar Parallax, as deduced from Tables 1 and 2.

Hour of Moon's transit.	Difference.
1	2 minutes.
2	1½ "
3	0¾ "
4	0 "
5	0 "
6	0½ "
7	1½ "
8	3 "
9	3 "
10	3 "
11	3 "
12	2½ "

Table 6.—Differences in Time of High Water for 20° Southern Solar Declination, as deduced from Tables 3 and 4.

Hour of Moon's transit.	Difference.
1	+1 minute.
2	+3 "
3	+5⅓ "
4	+8 "
5	+9 "
6	+9⅔ "
7	+8⅔ "
8	+3⅓ "
9	0 "
10	0 "
11	0 "
12	0 "

According to the rate given in Table 5, a lower parallax retards, and a higher one accelerates, the appearance of high water (see Tables 1 and 2), which is in conformity with the previous theoretical remarks on this subject.

The sign + prefixed to the differences in Table 6, means that 20° Southern solar declination belate the high water to that extent in these hours of lunar transit, as compared with the time of high water when the Sun stands in the equator; the same amount of

Northern solar declination would accelerate the arrival of high water in the same hour to the same extent. After the twelfth hour, the matter is reversed; in the fifteenth hour, for instance, 20° of Southern solar declination cause the high water to come 5⅓ minutes earlier.

Table 7 renders the means of Tables 3 and 4, and Table 8 renders these means as reduced to the mean lunar parallax of 57 minutes and the mean solar declination of 0°; that is, the equatorial position of the Sun, such reductions having been made on the basis of Tables 5 and 6.

Table 7.—Means of Tables 3 and 4.

Hour of Moon's transit.	Number of Observations.	Luni-tidal Interval.	Lunar Parallax.	Solar Declination.
		hrs. min.	minutes.	degrees.
1	91	8 21⅓	57'35	—0⅓
2	93	8 10½	57'15	—0¼
3	94	8 01	57'10	+0½
4	106	7 53½	56'90	—0½
5	96	7 55	56'60	—0¼
6	102	8 01¼	56'60	—0½
7	99	8 14	56'55	0
8	84	8 31¾	56'55	—0⅛
9	86	8 43	57'00	—0½
10	86	8 42	57'15	—0½
11	75	8 40¼	57'15	—0½
12	89	8 32½	57'20	0

Table 8.—Means of Tables 3 and 4, as Corrected according to Tables 5 and 6.

Hour of Moon's transit.	Number of Observations.	Luni-tidal Interval.	Lunar Parallax.	Solar Declination.
		hrs. min.	minutes.	degrees.
1	91	8 22	57	0
2	93	8 10¼	57	0
3	94	8 01	57	0
4	106	7 53¼	57	0
5	96	7 55	57	0
6	102	8 01¼	57	0
7	99	8 13¼	57	0
8	84	8 30¼	57	0
9	86	8 43	57	0
10	86	8 42½	57	0
11	75	8 40¾	57	0
12	89	8 33	57	0

Mean of 1,101 observations: 8 h, 19 m., which is the corrected establishment at Cape Henlopen.

According to Table 8, the mean curve of the semi-menstrual inequality of time of high water has been constructed as shown by the heavy full line in *Figs. 16* and *17*. The small circles on such line are the means as actually derived from Table 8, and it will be noticed that their simple connection renders a continuous curve, in one case only a departure of two minutes having been necessary to

secure continuity. The broken lines in the diagrams represent the influence of solar declination and of lunar parallax. *Fig. 16* has been extended over all the twenty-four hours, in order to show the reversed action of the solar declination in the last twelve hours.

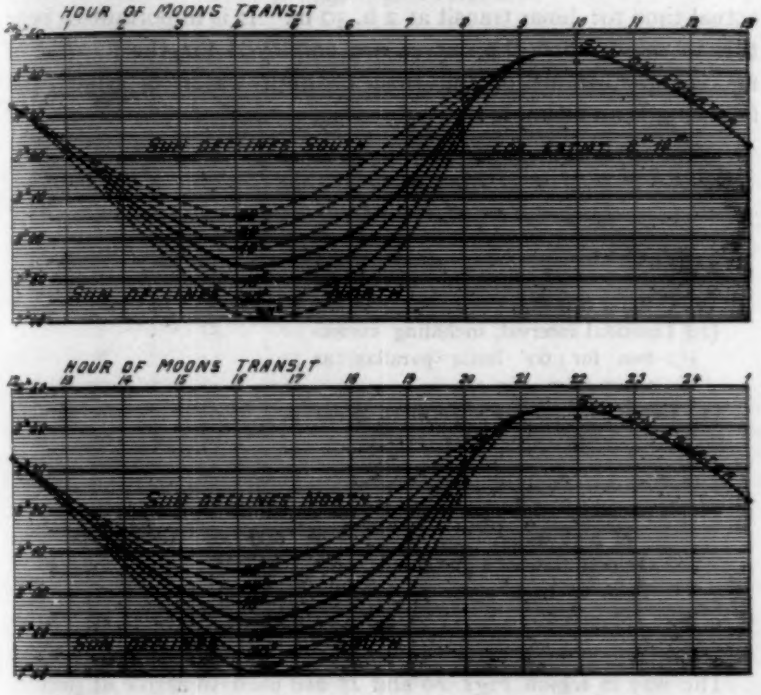

Fig. 16. The Curve of Semi-menstrual Inequality of the Time of High Water at Cape Henlopen, and the Influence of Solar Declination on it.

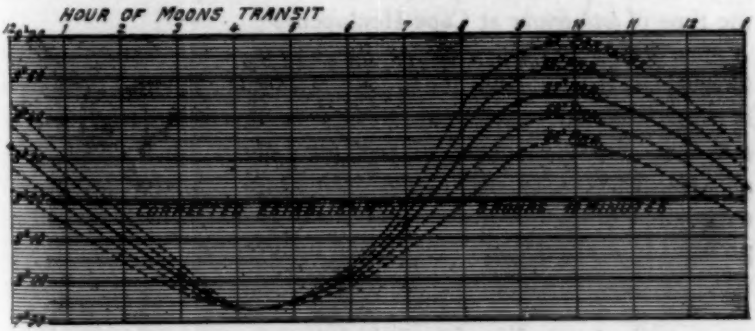

Fig. 17. The Curve of Semi-menstrual Inequality of the Time of High Water at Cape Henlopen, and the Influence of Lunar Parallax on it.

It should be mentioned here that the hour figures of lunar transit, as given in the diagrams, each represent the middle of such hour; thus, for instance, the figure 3 represents the middle of the third hour of lunar transit, which would be equivalent in actual time to: lunar transit at 2 h. 30 m. It is important to bear this in mind, when the curves are employed for the purpose of making predictions of the time of high water, of which the following is an example:

Prediction of the Time of High Water at Cape Henlopen on the Morning of May 27, 1884.

		Hours.	Minutes.	
(1.) Astronomical time of generating lunar transit, ...	May 26,	14	19½	
(2.) Lunitidal interval, including correction for 60′ lunar parallax, as taken from *Fig. 17*, ...		7	59	
(3.) Correction for 21° Northern solar declination (from *Fig. 16*), . .		+ 0	05	
Consequently, astronomical time of high water, . . .	May 26,	22	23½	
Or, civil time of high water, . .	May 27,	10	23½	A. M.
Time of high water as actually observed, ...	May 27,	10	15	A. M.

The way in which *Figs. 16* and *17* are used to arrive at predictions is so apparent from this example, that it does not seem necessary to add any further explanation, and we now proceed to discuss the time of *low water* at Cape Henlopen.

The following tables render the means of low water lunitidal intervals, as derived from the second reduction, which was grouped in three different ways, with a view to investigate the various influences, firstly, of lunar parallax, secondly, of lunar declination, and thirdly, of solar declination. The sign — prefixed to lunar declination always indicates first lunar group, and the sign + prefixed to it always indicates second lunar group:

The Influence of Lunar Parallax on the Time of Low Water at Cape Henlopen.

Table 9.—The Higher Parallaxes.

Hour of Moon's Transit.	Number of Observations.	Lunitidal Interval.	Lunar Parallax.	Lunar Declination.	Solar Declination.
		hrs. min.	minutes.	degrees	degrees.
1	52	14 19⅓	59·4	+0⅛	+0½
2	52	14 08⅝	59·2	+1¼	—0½
3	52	14 00½	59·0	+0⅛	0
4	47	14 00	58·8	+1	—1½
5	41	14 00	58·3	—1	—2⅓
6	44	14 09¼	58·3	—1⅓	—2
7	52	14 19	58·2	—0¼	—1½
8	44	14 32⅔	58·3	0	—2½
9	51	14 35⅓	58·9	—2⅛	+0½
10	46	14 41½	59·4	—1⅓	—1
11	47	14 29½	59·4	+0½	—2⅛
12	49	14 27¼	59·3	+1	+0¼

Table 10.—The Lower Parallaxes.

Hour of Moon's Transit.	Number of Observations.	Lunitidal Interval.	Lunar Parallax.	Lunar Declination.	Solar Declination.
		hrs. min.	minutes.	degrees.	degrees.
1	45	14 23	54·7	—0¼	—3
2	41	14 15	54·8	+2¾	+0¼
3	48	14 02	54·8	0	—0½
4	45	13 52	55 0	+0⅓	—1⅔
5	46	14 00¾	54·9	+1	+0⅛
6	52	14 08⅜	55·0	—1½	—2
7	52	14 18	54·9	—2⅛	+0⅓
8	46	14 39½	54·8	—1	—2⅓
9	47	14 45¼	54·8	—2¼	—1
10	49	14 48	54·8	—2½	+0¼
11	43	14 43	54·7	—2	+0⅓
12	38	14 34	54·8	—1¼	+2⅝

The Influence of Lunar Declination on the Time of Low Water at Cape Henlopen.

Table 11.—First Lunar Group.

Hour of Moon's Transit	Number of Observations.	Lunitidal Interval.	Lunar Parallax.	Lunar Declination.	Solar Declination.
		hrs. min.	minutes.	degrees.	degrees.
1	47	14 08¼	57·2	—15¼	—15
2	41	14 00	57·2	—15½	—12¾
3	47	13 50¾	57·0	—15¾	—12
4	47	13 42½	57·1	—14	— 8
5	42	13 52⅓	56·7	—15¼	— 4¾
6	50	13 58⅜	56·6	—14¾	— 2
7	58	14 09⅜	56·6	—13⅛	+ 3
8	47	14 23½	56·5	—13⅛	+ 5¼
9	54	14 29	57·0	—13¾	+ 9½
10	54	14 32⅓	57·0	—14	+13¾
11	49	14 26	57·1	—13⅛	+13
12	42	14 16½	57·2	—14½	+14½

Table 12.—Second Lunar Group.

Hour of Moon's Transit.	Number of Observations.	Lunitidal Interval.	Lunar Parallax.	Lunar Declination.	Solar Declination.
		hrs. min.	minutes.	degrees.	degrees.
1	50	14 31	57·4	+15	+12½
2	52	14 21¼	57·1	+16	+12
3	53	14 10⅜	57·2	+14½	+ 8½
4	45	14 10¼	56·8	+16	+ 6
5	45	14 08	56·5	+14	+ 5½
6	46	14 20	56·5	+13	0
7	46	14 29	56·6	+14	+ 0½
8	43	14 50	56·7	+13⅛	— 5
9	44	14 53¾	57·3	+12	— 9
10	41	14 56½	57·0	+14	—12
11	41	14 48¾	57·4	+14⅛	—13½
12	45	14 41¾	57·5	+14	—12½

The Influence of Solar Declination on the Time of Low Water at Cape Henlopen.

Table 13.—Northern Solar Declination.

Hour of Moon's Transit.	Number of Observations	Lunitidal Interval.	Lunar Parallax.	Lunar Declination.	Solar Declination.
		hrs. min.	minutes.	degrees.	degrees.
1	47	14 31¼	57·4	+15	+14
2	50	14 19⅜	57·0	+14	+13¾
3	49	14 10	57·0	+ 9	+15¾
4	44	14 03⅜	56·9	+ 8	+15
5	45	14 05	56·5	+ 3	+14½
6	47	14 14⅓	56·6	0	+14¾
7	56	14 18⅓	56·5	— 2½	+14½
8	49	14 29¼	56·6	— 5	+13½
9	54	14 30⅛	57·2	— 9	+15¾
10	59	14 34⅜	57·0	—11	+14
11	49	14 24½	56·9	—12½	+13½
12	46	14 16	57·2	—13	+13½

Table 14.—Southern Solar Declination.

Hour of Moon's Transit.	Number of Observations.	Lunitidal Interval.	Lunar Parallax.	Lunar Declination.	Solar Declination.
		hrs. min.	minutes.	degrees.	degrees.
1	50	14 09	57·3	—13½	—14½
2	43	14 02¾	57·3	—12	—14½
3	51	13 53¾	57·2	—10	—15
4	48	13 49½	56·9	— 6	—16
5	42	13 55½	56·7	— 3½	—15
6	49	14 03⅓	56·6	— 3	—16
7	48	14 18⅛	56·6	0	—14
8	41	14 44	56·5	+ 4½	—15
9	44	14 51½	56·8	+ 6	—14½
10	36	14 56	57·3	+13½	—15
11	41	14 49½	57·4	+13½	—14¾
12	41	14 44¾	57·2	+15	—13½

The determination of the influence of lunar parallax from Tables 9 and 10 does not present any difficulty, the other sidereal causes being practically equal in these two tables. In Tables 11 and 12, and in Tables 13 and 14, however, the influences of lunar and solar declination are blended together in most of the hours, and to obtain a result we have to resort to gradual approximation in

about the following way: A preliminary determination is made of the influence of lunar declination, using, for this purpose, only the fifth, sixth, seventh and eighth hour, during which solar declination in Table 11 is about the same as in Table 12. This preliminary determination renders three-fourths of a minute difference in time per 1° difference in lunar declination, and on the basis of this result the lunitidal intervals of Tables 13 and 14 are reduced to a lunar declination of 0°, thus to arrive at the influence of the solar declination. The values of this latter are then employed for correcting Tables 11 and 12, and thus, by gradual approximation, we finally arrive at these results:

Table 15.—Difference in Time of Low Water per Minute of Lunar Parallax as deduced from Tables 9 and 10.

Hour of Moon's transit.	Difference.
1	1 minute.
2	0¾ "
3	0½ "
4	0 "
5	0 "
6	0¼ "
7	1 "
8	1½ "
9	2 "
10	2½ "
11	2 "
12	1½ "

Table 16.—Difference in Time of Low Water for 20° Southern Solar Declination as deduced from Tables 13 and 14.

Hour of Moon's transit.	Difference.
1	0 minute.
2	—1½ "
3	—2½ "
4	—3½ "
5	—3½ "
6	—2½ "
7	0 "
8	+5 "
9	+7 "
10	+6 "
11	+3½ "
12	+1½ "

As to whether the figures in these tables indicate retardations or accelerations of the arrival of low water, the remarks that were made in this connection at the foot of Tables 5 and 6 also apply here.

For the influence of lunar declination, we obtain from the corrected figures of Tables 11 and 12 the result that 10° difference in declination will cause a difference of about $7\frac{1}{10}$ minutes in time, the question of retardation or acceleration here depending, as we know, on the lunar group.

Table 17 renders the means of Tables 11 and 12, and Table 18 renders these means as reduced to the mean lunar parallax of 57

minutes, and the mean lunar and solar declination of 0°; that is, the equatorial position of Sun and Moon, such reduction having been made on the basis of Tables 15 and 16 and the above given magnitude of influence of lunar declination.

Table 17.—Means of Tables 11 and 12.

Hour of Moon's transit.	Number of Observations.	Lunitidal Interval.	Lunar Parallax.	Lunar Declination.	Solar Declination.
		hrs. min.	minutes.	degrees.	degrees.
1	97	14 19½	57·30	0	−1¼
2	93	14 10½	57·15	+0¼	−0½
3	100	14 00¾	57·10	−0½	−1¾
4	92	13 56½	56·95	+1	−1
5	87	14 00	56·60	−0½	+0½
6	96	14 09½	56·55	−1	−1
7	104	14 19¼	56·60	+0½	+1¾
8	90	14 36¾	56·60	0	+0½
9	98	14 41½	57·15	−1	+0¼
10	95	14 44½	57·00	0	+1
11	90	14 37½	57·25	+0½	−0¼
12	87	14 29	57·35	−0¼	+1

Table 18.—Corrected Means of Tables 11 and 12.

Hour of Moon's transit.	Number of Observations.	Lunitidal Interval.	Lunar Parallax.	Lunar Declination.	Solar Declination.
		hrs. min.	minutes.	degrees.	degrees.
1	97	14 20	57·0	0	0
2	93	14 10½	57·0	0	0
3	100	14 01½	57·0	0	0
4	92	13 55¾	57·0	0	0
5	87	14 00½	57·0	0	0
6	96	14 10¼	57·0	0	0
7	104	14 18¼	57·0	0	0
8	90	14 36	57·0	0	0
9	98	14 42½	57·0	0	0
10	95	14 44¾	57·0	0	0
11	90	14 37½	57·0	0	0
12	87	14 29¾	57·0	0	0

Mean of 1,129 observations: 14 h. 20½ m., which is mean low water lunitidal interval at Cape Henlopen.

According to Table 18, the mean curve of the semi-menstrual inequality of the time of low water has been constructed as shown by the heavy full line in *Figs. 18* and *19*. The small circles on such line are the means as actually derived from Table 18, and it will be noticed in this case also, that the continuity of the curve necessitated only very trifling departures from the means as actually found. The broken lines in the diagrams represent the influence of lunar parallax and declination. The influence of the solar declination (Table 16) has not been represented, the results found for it not being considered very reliable, on account of the heavy corrections and following therefrom the indirectness of the way required to obtain them

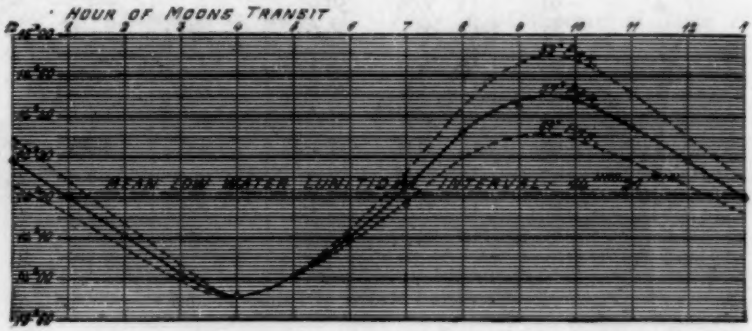

Fig. 18. The Curve of Semi-menstrual Inequality of the Time of Low Water at Cape Henlopen, and the Influence of Lunar Parallax on it.

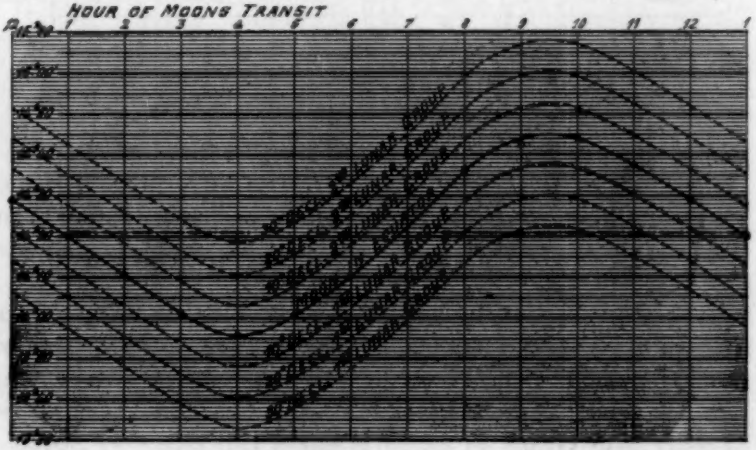

Fig. 19. The Curve of Semi-menstrual Inequality of the Time of Low Water at Cape Henlopen, and the Influence of Lunar Declination on it.

The following is an example of utilizing the curves of *Figs. 18* and *19* for the purpose of predicting the time of low water.

Prediction of the Time of Low Water at Cape Henlopen on the Afternoon of May 27, 1884.

		Hours.	Minutes.	
(1.) Astronomical time of generating lunar transit, ...	May 26,	14	19½	
(2.) Lunitidal interval, including correction for 60′ lunar parallax, as taken from *Fig. 18*, ...		14	01½	
(3.) Correction for 18° lunar declination, first group (from *Fig. 19*), . .		— 0	13	
Consequently, astronomical time of low water, ...	May 27,	4	08	
Or, civil time of low water, . .	May 27,	4	08	P. M.
Time of low water as actually observed, ...	May 27,	4	10	P. M.

Deducting the above found corrected establishment from the mean low water lunitidal interval, we find the mean duration of fall, and deducting this from the mean duration of half a lunar day, we obtain the mean duration of rise. These operations render

	Hours.	Minutes
Mean low water lunitidal interval at Cape Henlopen, .	14	20½
Corrected establishment at Cape Henlopen, . . .	8	19
Consequently mean duration of fall, ...	6	01½
And mean duration of rise, ...	6	23½

We now proceed to discuss the time of the event of high and of low water at various points along the Delaware and the Pennsylvania shore, above Cape Henlopen as far up as Bordentown.

The first efforts of the writer in this direction were made several years ago in this way, that all the simultaneous observations available at the various points were compared, and the time of propagation of high and of low water from one point to the other derived, all such times being finally referred to Cape Henlopen as the starting point. The results obtained in this way are embodied in a table, which has been published by Professor Henry Mitchell, on page 8 of Appendix, No. 18, to the Coast Survey Report of 1881.

The present results have been arrived at by first determining "corrected establishment" and "mean low water lunitidal interval" for the following points independently of each other, in the same way as determined above for Cape Henlopen:

Edgemoor,
Marcus Hook,
Fort Mifflin,
Five Mile Point.

Together with Cape Henlopen, five standard points were thus obtained, and each of the intermediate points was then referred to two or more of such standard points, with the exception of Collins Beach, where simultaneous observations, with only one standard point, were available. In this way a check on the correctness of the computations was obtained, which the above-mentioned earlier results do not possess. If, in addition to this, it is remembered that the number of observations employed at present aggregates about five times as many as the number available for the earlier work, then the writer may appear justified in claiming his present results to be not far from the truth, and certainly nearer to it than his earlier ones.

The following table renders the results obtained and also shows from how many observations each of them was found, thus giving a fair scale for the reliability of each result.

Table 19.—The Time of High and of Low Water at various Points on Delaware Bay and River.

No.	Name of Station.	Distances in Nautical Miles from Cape Henlopen.	Corrected Establishment.	Number of Observations.	Mean Low Water Lunitidal Interval.	Number of Observations.	Time of Propagation from Cape Henlopen.		Duration of	
							of High Water.	of Low Water.	Rise.	Fall.
1	Cape Henlopen, . .	0	hrs. min. 8 19	1101	hrs. min. 14 20½	1129	hrs. min. 0	hrs. min. 0	hrs. min. 6 23½	hrs. min. 6 01½
2	Collins Beach,	42·1	10 21½	21	17 09	20	2 02½	2 48½	5 37½	6 47½
3	Port Penn, .	49·4	10 52½	75	17 50	67	2 33½	3 29½	5 27½	6 57½
4	Ft. Delaware,	54·2	11 14	45	18 08½	38	2 55	3 48	5 20½	7 04½
5	New Castle, .	58·9	11 34	80	18 45	80	3 15	4 24½	5 14	7 11
6	Pigeon Point,	61·9	11 41½	59	18 56	64	3 22½	4 35½	5 10½	7 14½
7	Edgemoor, .	64·7	11 56½	384	19 16½	426	3 37½	4 56	5 05	7 20
8	Marcus Hook,	70·4	12 14½	362	19 40	339	3 55½	5 19½	4 59½	7 25½
9	Fort Mifflin, .	80·9	13 09¾	395	20 37½	400	4 50¾	6 17	4 57¾	7 27¾
10	Five Mile Pt.	92·1	14 12½	179	21 41½	161	5 53½	7 21	4 56	7 29
11	Bordentown,	112·4	16 19	44	23 44	56	8 00	9 23½	5 00	7 25

The results for Bordentown are apparently the least satisfactory ones. Instead of the duration of fall further increasing, as in reality

it most likely does, our table shows a slight decrease as compared with Five Mile Point. The reason probably is to be sought in the fact, that all observations available for Bordentown were made during the dry season of the year, when the fresh water flow from above was comparatively small. Future observations, extending through all seasons of the year, will in all likelihood render a longer low water lunitidal interval and consequently a longer duration of fall at this point.

If the times of propagation from Cape Henlopen, as given in Table 19, are laid off as ordinates, the distances from Cape Henlopen being the abscisses, then two lines are obtained (one for high and one for low water), which, although irregular, yet approach regularity to such a degree, that one might be tempted, by means of some alterations, to substitute continuous curves for them. But in the first place the alterations necessary for this, at such points as Marcus Hook and Fort Mifflin, would, in the writer's opinion, exceed the limit of possible error in the lunitidal intervals as determined above for these points, which limit he does not place any higher than two or three minutes. Again, there is no reason to believe that the line in question should be a continuous curve, but there is on the contrary much reason to believe that it cannot be one, as the following deduction may serve to show.

The below-named conditions may be said to determine the resistances which the tidal wave in its course upwards has to overcome, and consequently likewise to determine the velocity of its propagation:

(1.) Grade of river bottom.

(2.) Form and size of cross-sectional areas.

(3.) Curvature of river.

(4.) Fresh water flow.

(5.) Rise and fall of tide at mouth.

(6.) Distance of propagation.

The two last-named conditions are practically constants, and if for the four others we assume as follows:

(1.) Uniform grade throughout;

(2.) Uniform shape of cross-sections, and decrease of cross-sectional areas according to a continuous law;

(3.) Uniform curvature throughout;

(4.) Uniform fresh water flow throughout, coming down in the river bed only, no tributaries existing;

then the supposition would be justified that the velocity of propagation of high and low water would decrease, according to a continuous law, and that hence our above-described graphical representations of such velocity would assume the form of continuous curves.

None of the above-made suppositions, however, exist in reality. The tidal wave, therefore, in its course upwards, has to overcome resistances of a continually fluctuating character, each of them individually now increasing, now decreasing, and it would be a most remarkable and not-to-be-looked-for result if, under these circumstances, the fluctuations of the various resistances were at all places so to balance each other as to cause a decrease of velocity subject to a law expressible in a continuous curve. In the writer's opinion, therefore, the velocity of the tidal wave moving up any river is a quantity decreasing on the whole, but subject to local irregularities corresponding to the local conditions of the river bed. It seems quite possible that researches extended over a number of rivers may reveal the amount of influence exercised by each of the above-named conditions, and that thus we may be enabled, from a full knowledge of such conditions in any given case, to draw conclusions in regard to the propagation of the tidal wave.

Table 19 can be utilized for predicting the time of high and low water at any of the points contained in it in this way, that the prediction is first made for Cape Henlopen, as described before, and that then the time of propagation for the point in question, as taken from the table, is added. This time of propagation is, however, by no means a constant, but is influenced by the hour of lunar transit, by lunar declination, and by lunar parallax; and the attempt has been made, from the joint observations for Edgemoor, Marcus Hook and Fort Mifflin, to determine the magnitude of such influences for the region of these points.

The influence of lunar paralliax, although apparent, could not be determined with any accuracy.

The influence of lunar declination on the time of propagation of low water was found to amount to about two minutes per 5° of declination, the correction to be applied in the sense of the two lunar groups.

The influence of the hour of Moon's transit is apparent from the following two curves of semi-menstrual inequality.

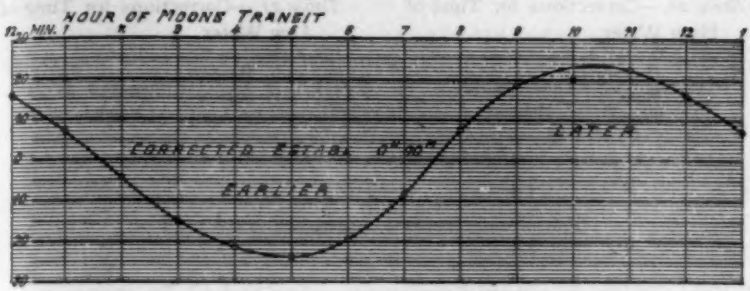

Fig. 20.—The Curve of Semi-menstrual Inequality of the Time of High Water in the Region of Edgemoor, Marcus Hook and Fort Mifflin, as deduced from 1,141 Observations.

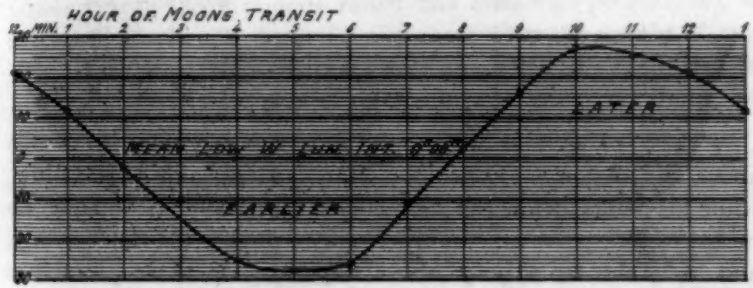

Fig. 21.—The Curve of Semi-menstrual Inequality of the Time of Low Water in the Region of Edgemoor, Marcus Hook and Fort Mifflin, as deduced from 1,165 Observations.

As indicated by the headings, these curves were obtained by averaging the means for each lunar hour of the above-named three points, thus rendering a result for some point (within the region named), the exact position of which is not known, and for which was found: Corrected establishment = 12 hours, 26⅔ minutes, and mean low water lunitidal interval = 19 hours, 51 minutes. The shape of the curves cannot, in the entire region, vary very much, and they may, consequently, with safety, be applied to any of the three points by simply substituting his corrected establishment in the case of high water, and his mean low water lunitidal interval in the case of low water.

From a comparison of these two curves with the ones found for Cape Henlopen, we deduce the following tables of corrections, to be applied to the times of propagation of high and low water (as taken from Table 19) in each hour of lunar transit.

Table 20.—Corrections for Time of High Water.

Hour of Moon's Transit.	Correction in Minutes.
1	+4
2	+4
3	+3
4	+4
5	0
6	—2
7	—3
8	—4
9	—5½
10	—2
11	+0½
12	+2

Table 21.—Corrections for Time of Low Water.

Hour of Moon's Transit.	Correction in Minutes.
1	+13
2	+ 8½
3	+ 5½
4	0
5	— 8½
6	—15¼
7	—13
8	—12
9	— 6½
10	+ 3¼
11	+11½
12	+12¼

These corrections will apply approximately also to points above the region of Edgemoor, Marcus Hook and Fort Mifflin, while for points below the corrections can easily be obtained by means of interpolation. In order now to predict the time of high and low water for a point above Cape Henlopen, we proceed as shown in the following example:

Prediction of Time of High and Low Water at Marcus Hook, on May 27, 1884.

	Hours.	Minutes.	
(1.) Time of high water at Cape Henlopen as found before,	10	23½	A. M.
(2.) Time of propagation (Table 19),	3	55½	
(3.) Correction for third lunar hour, (Table 20)	+	3	
Consequently time of high water at Marcus Hook,	2	22	P. M.
(1.) Time of low water at Cape Henlopen as found before,	4	10	P. M.
(2.) Time of propagation (Table 19),	5	19½	
(3.) Correction for 18° lunar declination, 1st group,	—	7	
(4.) Correction for third lunar hour, (Table 21)	+	5½	
Consequently, time of low water at Marcus Hook,	9	28	P. M.

(*To be concluded.*)

AN ACCOUNT OF THE EXPERIMENTS MADE ON A CONDENSING COMPOUND ENGINE, BY A COMMITTEE OF THE INDUSTRIAL SOCIETY OF MULHOUSE, IN ALSACE, GERMANY.

BY CHIEF ENGINEER ISHERWOOD, U. S. N.

In 1878, a medal of honor was offered by the Industrial Society of Mulhouse, in Alsace, Germany, for the first compound engine constructed in upper Alsace, that would give the French horse-power (32544·17077 pounds raised one foot high per minute), *as shown by a friction brake* for not more than the weight of nine kilograms (19·8416 pounds) of steam consumed per hour, equivalent to about 17·44 pounds of steam consumed per hour *per indicated English horse-power.*

In 1879, the above challenge was accepted by Messieurs C. Weyher and Richemond, agents of the Central Company for the construction of the Pantin engine, who offered for the prize one of their engines then in operation at the spinning factory of Mr. Antoine Herzog, at Colmar, Alsace, and which had been regularly working about fourteen months.

Accordingly, the Industrial Society directed its Committee on Mechanics, consisting of Messieurs Goerich, Grosseteste, X. Fluhr, Keller, Poupardin and Walter Meunier, to experiment with and report upon the engine offered. These gentlemen made several very careful tests, and most accurately ascertained the cost in pounds of feed water consumed per hour of the indicated horses-power developed in the cylinders, and of the horses-power delivered to the shafting of the factory as determined by a friction brake applied to the engine shaft; but they did not make the most of their opportunity and labor, by ascertaining the condensation of steam in the cylinders and other interesting facts, although the data in their possession would have enabled them to do so with very little additional work. They do not even state the measure of expansion with which the steam was used, nor at what point of the stroke of the pistons the steam was released from and cushioned in the cylinders, though these facts are essential to an

understanding of the causes producing the particular results obtained. They do, indeed, give a specimen indicator diagram from each end of the two cylinders for each experiment, from which the reader may ascertain approximately when the steam was cut off, released and cushioned, but they should have given the mean fraction of the stroke of the pistons from all the diagrams, at which these operations upon the steam took place. The back pressures against all the pistons should also have been given, both including and excluding the cushioning, and the back pressure against each piston at the point where the cushioning commenced should have been given. Likewise, the mean pressure of the expanded steam alone should have been given, and the pressure at the end of the stroke of each piston. The omissions mentioned could have readily been supplied from the diagrams in possession of the committee, and always should be in every case of experimenting with steam engines, as without this information only a very imperfect idea can be had of the more or less proper distribution of the pressures in the cylinders, and to what degree the economy of the steam may have thus been affected; a proper experiment and report ascertains and presents the *whole* subject, nothing is more unsatisfactory than partial data which may easily lead to erroneous conclusions. Had a drawing been given of the boiler, and had the coal been weighed that was consumed during the long experiments of October 22d and 23d, each of over twelve hours' duration, the data of a boiler experiment would have been completed, for all the other quantities are given, and the value of the investigation that much extended. It is true that the coal was weighed during the experiments of July 7th and 8th, but they were entirely too short for reliable vaporizations, being only from three to four hours long.

The report of the committee will be found in the *Bulletin of the Society*, for January and February, 1880, pages 5 to 20, in which are detailed the methods of observation and calculation employed, and the results obtained.

From this report, the following facts have been taken as far as given, and the remainder supplemented from the indicator diagrams above mentioned. No regard has been had to the arrangement, calculations, or inferences of the committee; their observed facts alone have been used.

The feed water was carefully measured in a tank before being pumped into the boiler. It was a portion of the water delivered from the jet condenser by the air-pumps.

The condensing water was also carefully measured in a tank before it entered the jet condenser; and the quantity of combined condensing water and water of steam condensation withdrawn from the condenser by the air-pumps, was likewise measured in another tank into which it was thrown by these pumps.

Indicator diagrams were taken every twenty minutes from each end of each cylinder, and from the mean pressure shown by all of them, the indicated horses-power were computed.

By a very excellent arrangement of friction brake applied to a drum on the engine-shaft, there were ascertained the number of horses-power delivered by the engine to the factory shafting. Of course, this power was less than the indicated power by the friction resistance of the engine *per se*, or unloaded, and by the friction of the load on the journals and guides of the working parts of the engine.

The entire work of the engine during the experiments of July 7 and 8, 1879, consisted in overcoming the friction produced by the brake. During each of these experiments, the steam pressure in the boiler was maintained almost without variation, the point at which the steam was cut off in the cylinders remained fixed, and the throttle valve was kept wide open, so that the mean of all the indicator diagrams taken must represent, with absolute accuracy, the mean performance of the engine.

The number of double strokes made by the pistons of the engine was taken from a counter; and, in the weighing and measuring of the several observed quantities, all the usual corrections were made. The indicators were also carefully measured and tested.

A sufficient number of qualified assistants were employed to secure absolute accuracy in the quantities observed.

The data and result of these experiments, made in July, 1879, will be found in the following, Table No. 1. In the succeeding October, the committee made two additional trials on the engine while doing its regular work at the factory, each of which continued a little over twelve hours, in order to test the accuracy of the July experiments, and to ascertain whether the engine when functioning under the conditions of ordinary practice gave the same

economic returns as when tried under the previous experimental conditions. The results of the October tests will be found in the following Table No. 2, which contains all the data taken.

Before proceeding to an examination of the tables, it is necessary to have the description and dimensions of the engine, given below:

ENGINE.

The engine is of the portable kind; that is, it is secured upon the top of its boiler, the latter being cylindrical and horizontal, with lugs attached by which it can be placed temporarily on any foundation and readily removed from place to place.

The engine works with condensation, and consists of two compounded cylinders of unequal diameters and equal strokes of pistons, lying horizontally side by side and bolted to a strong cast iron bed-plate, which in turn is bolted to the top of the boiler. The cylinders are direct acting, with piston rods secured into crossheads working between guides, from which crossheads connecting rods extend to the crank-pins of the shaft. The shaft is horizontal and curved out into cranks opposite the respective cylinders. It is supported by three pillar-blocks, one at each end outside the cranks, and the third midway between the two. One of the projecting ends of the shaft carries the fly-wheel. The eccentrics for operating the valves of both cylinders are keyed at proper positions upon the shaft. The valve chest of each cylinder is upon its outer side.

The steam valve of the small cylinder is the usual short slide operated by the usual eccentric. In addition to the steam valve, there is an independent slide cut-off valve operated by an eccentric, and capable, through suitable mechanism, of varying the point at which the steam is cut off by means of the usual governor acting upon a double cam. When the engine is in operation, the point of cutting off automatically varies with variations of the load.

The large cylinder has no independent cut-off valve; its steam valve is the usual short slide operated by the usual eccentric. The steam is cut off, however, at an invariable point in the large cylinder, by means of lap on the steam side of the steam valve. The valves of neither cylinder were counterbalanced, but worked with the full valve chest pressure upon their backs.

As the load of the engine during the experiments of July 7th and 8th, hereinafter described, consisted entirely of friction produced by a brake carrying the weight always at the same leverage, which weight was the same for all the experiments except that of the morning of July 8th, in which it was reduced nearly one-half, the point of cutting off in the small cylinder remained invariable throughout each experiment. For the experiment of the morning of July 8th, the steam was cut off in the small cylinder at 0·25 of its stroke from the commencement, and for the remaining experiments at 0·42 of its stroke. The steam in all the experiments was released from the small cylinder when 0·98 of the stroke of its piston was completed, and cushioned when 0·925 of the return stroke was completed.

The point of cutting in the large cylinder, during the abovementioned experiments, was when 0·45 of the stroke of the piston was completed. The steam was invariably released from the large cylinder when 0·91 of the stroke of its piston was completed, and cushioned when 0·75 of the return stroke was completed.

The small cylinder has a steam jacket covering its entire cylindrical surface, the jacket being interposed between the cylinder and the valve chest. By this arrangement, the bottom of the valve chest becomes steam jacketed, which is advantageous to the economy, but the steam passages are lengthened by the width of the jacket, which is disadvantageous to a much greater degree. The ends of the cylinder are not jacketed.

The exhaust steam from the small cylinder contours the upper part of the outside of the jacket of that cylinder, taking heat from the jacket, and is delivered to the steam valve of the large cylinder after traversing a passage or jacket contouring the upper part of the large cylinder to its valve chest. The lower part of the large cylinder, excepting the surface covered by its valve chest, has a steam jacket which receives its steam directly from the boiler. Only one-half of the cylindrical surface of the large cylinder is jacketed with steam taken immediately from the boiler. The remainder of this surface is covered by the valve chest and by the exhaust passage leading to it from the small cylinder. The valve chest of the large cylinder is cast upon the cylinder itself, the side of which forms the bottom of the chest, so that no jacket space interposes between them as in the case of the small cylinder. Neither end of the large cylinder is jacketed.

There is no separate steam pipe leading from the boiler to the jackets of the cylinders. The main steam pipe delivers the boiler steam into the jacket of the small cylinder a little above the lowest point of that jacket. This pipe has an inclination downwards to the boiler, made in the expectation that the water of condensation from the jackets would drain back by gravity to the boiler. From the jacket, the steam enters the small cylinder.

The steam jacket of the large cylinder receiving its steam directly from the boiler, is supplied through a short branch from the main steam pipe, which branch enters that jacket a little above its lowest point.

To resume: Both the heads of both cylinders are not steam jacketed. The entire cylindrical surface of the small cylinder and one-half of this surface of the large cylinder are steam jacketed. The remaining half of the cylindrical surface of the large cylinder is covered by its valve chest and by the exhaust passage leading to it from the small cylinder.

The exterior surfaces of the cylinders, with the exception of the heads, are clad with non-heat-conducting substances.

There are two air-pumps worked by a T lever actuated by a short connecting rod articulated to the crosshead. These pumps are vertical and immersed in a tank. The feed-pump is articulated to one of the arms of the lever working the air-pumps.

There is one jet condenser of very great capacity relatively to the volume of the large cylinder. The injection water is sprayed into it by a conical rose.

The speed of the engine is automatically regulated by a Porter governor acting on the cut-off cam by means of the Denis compensator, an apparatus that Messieurs Weyher and Richemond apply to nearly all their engines with excellent results.

The following are the dimensions of the engine:

Diameter of the small cylinder, . . .	11·2010	inches.
Diameter of the piston rod of the small cylinder, ...	2·1654	inches.
Stroke of the piston of the small cylinder,	18·8980	inches.
Net area of the small cylinder piston, .	96·696816	square inches.
Space displacement of the small cylinder piston, per stroke, ...	1·0575095	cubic feet.
Space in clearance and steam passage at one end of small cylinder, . .	0·0676805	cubic foot.

Per centum which the space in clearance and steam passage at one end of the small cylinder is of the space displacement of its piston, per stroke,	6·40	
Length of the steam port of the small cylinder,	4·7245	inches.
Breadth of the steam port of the small cylinder,	0·9843	inches.
Area of the steam port of the small cylinder,	4·6503	square inches.
Diameter of the steam pipe,	3·1497	inches.
Depth of the piston of the small cylinder,	4·7245	inches.
Diameter of the large cylinder,	18·9059	inches.
Diameter of the piston rod of the large cylinder,	2·1654	inches.
Stroke of the piston of the large cylinder,	18·8980	inches.
Net area of the large cylinder piston,	278·886568	square inches.
Space displacement of the large cylinder piston, per stroke,	3·0499990	cubic feet.
Space in clearance and steam passage at one end of large cylinder,	0·1525010	cubic foot.
Per centum which the space in clearance and steam passage at one end of the large cylinder is of the space displacement of its piston, per stroke,	5·00	
Length of steam port of large cylinder,	9·4490	inches.
Breadth of steam port of large cylinder,	0·9843	inch.
Area of steam port of large cylinder,	9·3006	square inches.
Diameter of the exhaust pipe,	4·5000	inches.
Depth of the piston of the large cylinder,	4·7245	inches.
Distance between the axes of the two cylinders,	23·6225	inches.
Thickness of metal of the 2 cylinders,	0·9843	inch.
Area of the piston of the large cylinder relatively to the area of the piston of the small cylinder taken as unity,	2·884134	
Aggregate space displacement of the piston of the large cylinder per stroke, and space in clearance and steam passage at one end of that cylinder, relatively to the aggregate space displacement of the piston of the small cylinder per stroke, and space in clearance and steam passage at one end of that cylinder taken as unity,	2·846186	

Table No. 1 contains all the observed quantities, and the calculated results from them, obtained during the special July trials of the engine. These quantities and results are arranged in groups for facility of reference. Each quantity and each result is so fully described on the line bearing it, that only in a few cases is any further explanation necessary.

The experiments were four in number, and were made with nearly the same piston speed in all, the difference being too small to exercise any practical effect on the economic result The boiler pressure and the condenser pressure also varied but very little during the different experiments, and the throttle valve was kept wide open in all, so that as regards these important conditions, all the experiments had equality, being made with equal boiler pressure, equal piston speed, equal throttle valve opening, and equal condenser pressure.

During all the experiments, except that of the morning of July 8th, the distribution of the steam was exactly the same, being cut off in the small cylinder when 0·42 of the stroke of its piston was completed, and in the large cylinder when 0·45 of the stroke of its piston was completed, the measure of expansion with which the steam was used being, by volume, 6·2569 times in all three experiments. For the small cylinder, the steam was released when 0·98 of the stroke of its piston was completed, and for the large cylinder when 0·91 of the stroke of its piston was completed. In the small cylinder, the cushioning of the back pressure steam commenced when 0·925 of the return or exhaust stroke of the piston was completed, and in the large cylinder when 0·75 of that stroke of its piston was completed. The three experiments, therefore, of the morning and afternoon of July 7th, and the afternoon of July 8th, were repetitional in order to secure certainty of the result on which the award of the prize was to depend.

In the single experiment of the morning of July 8th, the distribution of the steam in the large cylinder was exactly the same as in the case of the three other experiments; but, in the small cylinder, the steam was cut off when 0·25 of the stroke of its piston was completed, the release and the cushioning being the same as in the three other experiments. Thus the only difference was the changing of the point of cut-off in the small cylinder from

Table No. 1.—Containing the Data and Results of the Experiments made in July, 1879, on a Condensing Compound Engine in the Factory of Antoine Herzog, at Colmar, in Alsace, Germany, by the Committee on Mechanics of the Industrial Society of Mulhouse.

	July 7, 1879. Morning.	July 7, 1879. Afternoon.	July 8, 1879. Morning.	July 8, 1879. Afternoon.
Economi	Number of Fahrenheit units of heat consumed per hour per indicated horse-power, . . .	[illegible]	[illegible]	[illegible]
	Number of Fahrenheit units of heat consumed per hour per net horse-power, ...	21710·3·9960	20951·7·1781	21100·310264	20490·193161
	Number of Fahrenheit units of heat consumed per hour per total horse-power developed by the engine, ...	15178·954446	15327·7374·6	15142·576895	14965·219982
	Number of Fahrenheit units of heat consumed per hour per horse-power developed by	21785·130514	2156..·619794	21906·148580	21804·449220

Steam Cylinders.	Net pressure on the piston of the small cylinder, in pounds per square inch,	26·572993	27·179157	24·227924	27·046174
	Total pressure on the piston of the small cylinder, in pounds per square inch,	71·825000	73·200000	60·200000	73·300000
Steam Pressures in Large Cylinder per Indicator.	Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero,	43·000000	43·150000	34·490000	43·[illegible]
	Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero,	27·540000	27·670000	21·156000	27·740000
	Pressure on piston of large cylinder at the end of its stroke, in pounds per square inch above zero,	12·470000	12·529000	9·600000	12·561000
	Mean back pressure against piston of large cylinder during its stroke, in pounds per square inch above zero,	3·756000	3·756000	3·756000	3·756000
	Back pressure against piston of large cylinder at the point where the cushioning began, in pounds per square inch above zero,	2·355000	2·355000	2·355000	2·355000
	Indicated pressure on the piston of the large cylinder, in pounds per square inch,	22·100000	22·260000	17·006220	22·300000
	Net pressure on the piston of the large cylinder, in pounds per square inch,	20·841053	21·001053	15·747273	21·041053
	Total pressure, in pounds per square inch above zero, on the annular surface of the piston of the large cylinder remaining after subtracting from the area of that piston the area of the piston of the small cylinder,	24·650000	24·810000	19·556220	24·850000
Horses-Power.	Indicated horses-power developed in the small cylinder,	24·807987	25·168943	23·135964	25·113672
	Net horses-power developed in the small cylinder,	21·828234	22·205320	20·123093	22·143736
	Total horses-power developed in the small cylinder,	59·002239	59·804261	50·000578	60·013510
	Total horses-power developed in the small cylinder by the expanded steam alone,	27·966070	28·349834	28·101654	28·560045
	Indicated horses-power developed in the large cylinder,	52·358337	52·451950	40·738193	52·658120
	Net horses-power developed in the large cylinder,	49·375696	49·485453	37·722401	49·685305
	Total horses-power developed in the large cylinder by the annular surface of its piston remaining after subtracting from the area of that piston the area of the piston of the small cylinder,	38·151081	38·190874	30·603792	38·333920
	Total horses-power developed in the large cylinder by the expanded steam alone,	40·017681	40·047311	32·491077	40·194311
	Aggregate indicated horses-power developed by the engine,	77·166324	77·620893	63·874157	77·771792
	Aggregate net horses-power developed by the engine,	71·203930	71·690773	57·845494	71·829041
	Aggregate total horses-power developed by the engine, exclusive of cushioning,	97·151320	97·995135	80·604370	98·347430
	Aggregate total horses-power developed by the expanded steam alone in both cylinders, inclusive of cushioning,	67·983751	68·397145	60·592731	68·754356
	Horses-power developed by the engine at the friction brake applied to the wheel on the engine shaft,	67·690889	67·524224	55·717593	67·499569
	Per centum of the total horses-power developed by the engine utilized as indicated horses-power,	79·42900	79·20892	79·24403	79·07862
	Per centum of the total horses-power developed by the engine utilized as net horses-power,	73·29177	73·15748	71·76471	73·03601
	Per centum of the total horses-power developed by the engine utilized as horses-power at the friction brake,	69·67572	68·90569	69·12477	68·63379
Economic Results.	Number of pounds of feed water consumed per hour per indicated horse power,	17·071971	17·318·39	16·937819	16·920179
	Number of pounds of feed water consumed per hour per net horse-power,	18·521524	18·750553	18·703080	18·320085
	Number of pounds of feed water consumed per hour per total horse-power developed by the engine,	13·560c96	13·717432	13·422212	13·380260
	Number of pounds of feed water consumed per hour per horse-power developed by the engine at the brake,	19·461721	19·907547	19·417366	19·495149
	Number of Fahrenheit units of heat consumed per hour per indicated horse-power,	19110·090e3	19351·023136	19108·790286	18924·482602
	Number of Fahrenheit units of heat consumed per hour per net horse-power,	20710·3·9960	20951·7·1781	21100·310264	20490·193161
	Number of Fahrenheit units of heat consumed per hour per total horse-power developed by the engine,	15178·954446	15327·7374·6	15142·576895	14965·219982
	Number of Fahrenheit units of heat consumed per hour per horse-power developed by the engine at the brake,	21785·139514	21956··619794	21906·148580	21804·449220
Weight of Steam accounted for by the Indicator.	Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam, calculated from the pressure there,	987·257750	998·747724	607·871477	1005·726509
	Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, calculated from the pressure there,	1148·026328	1162·498768	879·050774	1170·906554
	Number of pounds of steam condensed per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed by the expanded steam alone in that cylinder,	77·162184	76·298208	76·530424	78·900853
	Sum of the two immediately preceding quantities,	1225·188512	1240·796976	955·581198	1249·807407
	Number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston, calculated from the pressure there,	1043·182517	1042·551556	805·995933	1047·498116
	Number of pounds of steam condensed per hour in the small and large cylinders, and in the receiver to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut-off valve on the small cylinder,	183·285887	184·699591	162·065479	185·699642
	Sum of the two immediately preceding quantities,	1226·468404	1227·251147	968·061412	1233·197758
Difference between the Weight of Water Vaporized in the Boiler and the Weight of Steam accounted for by the Indicator.	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam,	33·123471	345·493914	474·017445	310·187670
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam,	25 059069	25 721771	43·813874	23·572029
	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston,	92·192709	103·444662	126 307724	66·106772
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston,	6·99818	7 695392	11·674740	5·023638
	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston,	90·912817	116·990491	113·827510	82·716421
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston,	6·901·26	8·703085	10·521183	6·285852
Cylinder Pressures Reduced to Large Cylinder alone.	Indicated pressure, in pounds per square inch, that would be on the piston of the large cylinder were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder, and the quotient added to the experimental indicated pressure on the piston of the large cylinder,	32·571236	32·941408	26·664363	32·935300
	Net pressure in pounds per square inch, that would be on the piston of the large cylinder, similarly calculated to the immediately above,	30·054561	30·424733	24·147684	30·418625
	Total pressure, in pounds per square inch above zero, that would be on the piston of the large cylinder, corresponding to the total horses-power developed by the engine,	41·006728	41·587998	33·648414	41·648801
	Back pressure, in pounds per square inch above zero, that would be against the piston of the large cylinder similarly calculated to the above,	8·435492	8·646590	6 934051	8 713501

0·42 to 0·25 of the stroke of its piston from the commencement, whereby the measure of expansion with which the steam was used was made 9.6444 times by volume. The experiments, therefore, enable the determination to be made under equal and unexceptional conditions, and with extreme accuracy, of the economic results due to working high pressure steam in the partially steam jacketed cylinders of a condensing compound engine with pistons moving at a high reciprocating speed, with the measures of expansion 6·2569 and 9·6444 times, by volume. The results are very valuable and far reaching in engineering, and engineers should give them careful consideration and frank acceptance.

The number of pounds of feed water pumped into the boiler, the number of pounds of condensing water admitted to the condenser, and the number of pounds of condensing water and water of condensation withdrawn from the condenser, were obtained by separate measurements in different vessels, and are therefore three independent determinations. The last measurement should give the sum of the two previous ones, and it does so very nearly, the difference being of no practical importance. The very close approximation, however, shows the extreme care taken in the measurements.

The water data of the experiments afford a striking illustration of the transmutation of heat into power. The temperatures of the feed water, and of the condensing water when admitted into the condenser, being known, and the respective weights, the number of units of heat imported to the feed water in the boiler, and the number present in the condensing water and water of condensation when withdrawn from the condenser, are easily calculated, and the difference is the number of units of heat transmuted into the work done in the two cylinders of the engine by the *expanding* steam; that is, into the work done in those cylinders *after* the closing of the cut-off valve of the small cylinder.

An explanation is necessary of the manner in which the horses-power developed by the engine are calculated. Under the head of "horses-power" five kinds will be found in the table, namely, indicated, net, developed by the expended steam alone, total, and delivered by the engine to the factory shafting at the friction brake. The first three and the last are different fractions of the fourth.

Table No. 1.—Containing the Data and Results of the Experiments made in July, 1879, on a Condensing Compound Engine in the Factory of Antoine Herzog, at Colmar, in Alsace, Germany, by the Committee on Mechanics of the Industrial Society of Mulhouse.

		July 7, 1879. Morning.	July 7, 1879. Afternoon.	July 8, 1879. Morning.	July 8, 1879. Afternoon.
Total Quantities.	Duration of the experiments in consecutive hours and decimals of an hour,	2·99889	4 00139	3·00822	3·24139
	Total number of double strokes made by the pistons of the engine,	16014·	21253·	16211·	17253·
	Total number of pounds of feed water pumped into the boiler,	3950·681370	5378 835048	3248·068561	4265·391060
	Total number of pounds of condensing water admitted to the condenser,	73821·492916	98621·255129	74038·567197	78893·841895
	Total number of pounds of condensing water and water of steam condensation withdrawn from the condenser,	77872·440202	104133·100378	77253·139183	84499·777450
Engine.	Steam pressure in boiler in pounds per square inch above the atmosphere,	91·780	91·524	91·694	92·164
	Pressure in the condenser in pounds per square inch above zero,	. . . . .	. . . . .	1·810	1·907
	Number of double strokes made per minute by the pistons of the engine,	89 005187	88·503429	89 994448	88·711972
	Position of the throttle valve,	Wide open.	Wide open.	Wide open.	Wide open
	Fraction completed of the stroke of the piston of the small cylinder when the steam was cut off,	0 42	0·42	0 25	0·42
	Fraction completed of the stroke of the piston of the small cylinder when the steam was released,	0·98	0·98	0·98	0·98
	Fraction completed of the stroke of the piston of the small cylinder when the steam was cushioned,	0 925	0·925	0·925	0·925
	Fraction completed of the stroke of the piston of the large cylinder when the steam was cut off,	0·45	0 45	0·45	0 45
	Fraction completed of the stroke of the piston of the large cylinder when the steam was released,	0·91	0 91	0·91	0·91
	Fraction completed of the stroke of the piston of the large cylinder when the steam was cushioned,	0 75	0 75	0·75	0·75
	Number of times the steam was expanded,	6·2569	6·2469	9 6444	6·2569
	Atmospheric pressure in pounds per square inch above zero,	14·222813	14 222813	14 222813	14·222813
Water.	Number of pounds of feed water pumped into the boiler per hour,	1317·381221	1344·241638	1081·888922	1315·914179
	Number of pounds of condensing water admitted to the condenser per hour,	24616·272326	24646·749012	24661·273057	24648 018873
	Temperature in degrees Fahrenheit of the condensing water when admitted to the condenser,	48·256	48·056	48·001	48·010
	Number of pounds of condensing water and water of steam condensation withdrawn from the condenser per hour,	25967·087890	26024·231674	25732·004711	26068 994305
	Temperature in degrees Fahrenheit of the condensing water and water of steam condensation when taken from condenser,	95·720	97·655	86·932	96·728
	Number of Fahrenheit units of heat consumed per hour,	1474655·460688	1502043 696320	1220557·870799	1471790·924665
Small Indicator.	Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero,	100·768628	102·475366	99·787254	102·966053
	Pressure on piston of small cylinder at point of cutting off the steam, in pounds per				

Water from the condenser per hour,	23967·087890	26024·231674	25732·004711	26068·994305
Temperature in degrees Fahrenheit of the condensing water and water of steam condensation when taken from condenser,	95·720	97·655	86·932	96·728
Number of Fahrenheit units of heat consumed per hour,	1474655·460688	1502043·696320	1220557·870799	1471790·924665
Steam Pressures in Small Cylinder per Indicator.				
Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero,	100·768628	102·475366	99·787254	102·966053
Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero,	89·262373	90·969111	88·280999	91·459798
Pressure on piston of small cylinder at the end of its stroke, in pounds per square inch above zero,	44·098469	44·941652	33·423600	45·184067
Mean back pressure against piston of small cylinder during its stroke, in pounds per square inch above zero,	45·333226	46·200018	35·343690	46·449021
Back pressure against piston of small cylinder at the point where the cushioning began, in pounds per square inch above zero,	41·590000	42·385000	34·846000	42·614000
Indicated pressure on the piston of the small cylinder, in pounds per square inch,	30·200448	30·806612	27·855379	30·673629
Net pressure on the piston of the small cylinder, in pounds per square inch,	26·572993	27·179157	24·227924	27·046174
Total pressure on the piston of the small cylinder, in pounds per square inch,	71·823000	73·200000	60·200000	73·300000
Steam Pressures in Large Cylinder per Indicator.				
Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero,	43·000000	43·150000	34·490000	43·200000
Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero,	27·540000	27·670000	21·136000	27·740000
Pressure on piston of large cylinder at the end of its stroke, in pounds per square inch above zero,	12·470000	12·529000	9·600000	12·561000
Mean back pressure against piston of large cylinder during its stroke, in pounds per square inch above zero,	3·756000	3·756000	3·756000	3·756000
Back pressure against piston of large cylinder at the point where the cushioning began, in pounds per square inch above zero,	2·355000	2·355000	2·355000	2·355000
Indicated pressure on the piston of the large cylinder, in pounds per square inch,	22·100000	22·260000	17·006220	22·300000
Net pressure on the piston of the large cylinder, in pounds per square inch,	20·841053	21·001053	15·747273	21·041053
Total pressure, in pounds per square inch above zero, on the annular surface of the piston of the large cylinder remaining after subtracting from the area of that piston the area of the piston of the small cylinder,	24·650000	24·810000	19·556220	24·850000
Horses-Power.				
Indicated horses-power developed in the small cylinder,	24·807987	25·168943	23·135964	25·113672
Net horses-power developed in the small cylinder,	21·828234	22·205320	20·123093	22·143736
Total horses-power developed in the small cylinder,	59·000439	59·804261	50·002578	60·013510
Total horses-power developed in the small cylinder by the expanded steam alone,	27·966070	28·349834	28·101654	28·566045
Indicated horses-power developed in the large cylinder,	52·358337	52·451950	40·738193	52·658120
Net horses-power developed in the large cylinder,	49·375096	49·485453	37·722401	49·685305
Total horses-power developed in the large cylinder by the annular surface of its piston remaining after subtracting from the area of that piston the area of the piston of the small cylinder,	38·151081	38·190874	30·603792	38·333920
Total horses-power developed in the large cylinder by the expanded steam alone,	40·017681	40·047311	32·491077	40·194311
Aggregate indicated horses-power developed by the engine,	77·166324	77·620893	63·874157	77·771792
Aggregate net horses-power developed by the engine,	71·203930	71·690773	57·845494	71·829041
Aggregate total horses-power developed by the engine, exclusive of cushioning,	97·151320	97·995135	80·604370	98·347430
Aggregate total horses-power developed by the expanded steam alone in both cylinders, inclusive of cushioning,	67·983751	68·397145	60·592731	68·754356
Horses-power developed by the engine at the friction brake applied to the wheel on the engine shaft,	67·690889	67·524224	55·717593	67·499569
Per centum of the total horses-power developed by the engine utilized as indicated horses-power,	79·42900	79·20892	79·24403	79·07862
Per centum of the total horses-power developed by the engine utilized as net horses-power,	73·29177	73·15748	71·76471	73·03601
Per centum of the total horses-power developed by the engine utilized as horses-power at the friction brake,	69·67572	68·90569	69·12477	68·63379
Economic Results.				
Number of pounds of feed water consumed per hour per indicated horse power,	17·071971	17·318·39	16·937819	16·920179
Number of pounds of feed water consumed per hour per net horse-power,	18·521524	18·750553	18·703080	18·320085
Number of pounds of feed water consumed per hour per total horse-power developed by the engine,	13·360096	13·717432	13·422212	13·380260
Number of pounds of feed water consumed per hour per horse-power developed by the engine at the brake,	19·461721	19·907547	19·417366	19·495149
Number of Fahrenheit units of heat consumed per hour per indicated horse-power,	19110·090823	19351·123136	19108·790286	18924·482602
Number of Fahrenheit units of heat consumed per hour, per net horse-power,	22710·206870	23051·743781	21100·210264	20490·103161

	the engine at the brake,	21785·139514	2156·619794	21906·146560	21004·449220
Weight of Steam accounted for by the Indicator.	Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam, calculated from the pressure there,	987·257750	998·747724	607·871477	1005·726509
	Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, calculated from the pressure there,	1148·026328	1162·498768	879·050774	1170·906554
	Number of pounds of steam condensed per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed by the expanded steam alone in that cylinder,	77·162184	78·298208	76·530424	78·900853
	Sum of the two immediately preceding quantities,	1225·188512	1240 796976	955·581198	1249·807407
	Number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston, calculated from the pressure there,	1043·182517	1042·551556	805·995933	1047·498116
	Number of pounds of steam condensed per hour in the small and large cylinders, and in the receiver to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut-off valve on the small cylinder,	183·285887	184 699591	162·065479	185·699642
	Sum of the two immediately preceding quantities,	1226 468404	1227 251147	968 061412	1233·197758
Difference between the Weight of Water Vaporized in the Boiler and the Weight of Steam accounted for by the Indicator.	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam,	33·123471	345·493914	474·017445	310·187670
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam,	25 059069	25 701771	43·813874	23·572029
	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston,	92·192709	103·444662	126 307724	66·106772
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston,	6·998182	7 695392	11·674740	5·023638
	Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston,	90·912817	116·990491	113·827510	82·716421
	Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston,	6·901126	8·703085	10·521183	6·285852
Cylinder Pressures Reduced to Large Cylinder alone.	Indicated pressure, in pounds per square inch, that would be on the piston of the large cylinder were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder, and the quotient added to the experimental indicated pressure on the piston of the large cylinder,	32·571236	32·941408	26·664363	32·935300
	Net pressure in pounds per square inch, that would be on the piston of the large cylinder, similarly calculated to the immediately above,	30·054561	30·424733	24·147684	30·418625
	Total pressure, in pounds per square inch above zero, that would be on the piston of the large cylinder, corresponding to the total horses-power developed by the engine,	41·006728	41·587998	33·648414	41·648801
	Back pressure, in pounds per square inch above zero, that would be against the piston of the large cylinder similarly calculated to the above,	8·435492	8·646590	6 934051	8 713501

0·42 to 0·25 of the stroke of its piston from the commencement, whereby the measure of expansion with which the steam was used was made 9.6444 times by volume. The experiments, therefore, enable the determination to be made under equal and unexceptional conditions, and with extreme accuracy, of the economic results due to working high pressure steam in the partially steam jacketed cylinders of a condensing compound engine with pistons moving at a high reciprocating speed, with the measures of expansion 6·2569 and 9·6444 times, by volume. The results are very valuable and far reaching in engineering, and engineers should give them careful consideration and frank acceptance.

The number of pounds of feed water pumped into the boiler, the number of pounds of condensing water admitted to the condenser, and the number of pounds of condensing water and water of condensation withdrawn from the condenser, were obtained by separate measurements in different vessels, and are therefore three independent determinations. The last measurement should give the sum of the two previous ones, and it does so very nearly, the difference being of no practical importance. The very close approximation, however, shows the extreme care taken in the measurements.

The water data of the experiments afford a striking illustration of the transmutation of heat into power. The temperatures of the feed water, and of the condensing water when admitted into the condenser, being known, and the respective weights, the number of units of heat imported to the feed water in the boiler, and the number present in the condensing water and water of condensation when withdrawn from the condenser, are easily calculated, and the difference is the number of units of heat transmuted into the work done in the two cylinders of the engine by the *expanding* steam; that is, into the work done in those cylinders *after* the closing of the cut-off valve of the small cylinder.

An explanation is necessary of the manner in which the horses-power developed by the engine are calculated. Under the head of "horses-power" five kinds will be found in the table, namely, indicated, net, developed by the expended steam alone, total, and delivered by the engine to the factory shafting at the friction brake. The first three and the last are different fractions of the fourth.

The indicated horse-power is computed from the mean indicated pressure given by the indicator diagrams taken from the cylinders, and is given separately for the small and large cylinders, the aggregate being the indicated horses-power developed by the engine.

The main shaft of the engine being disconnected from the factory shafting, and the engine worked *per se*, or unloaded, indicator diagrams were taken at different speeds of piston, and showed that the pressure per square inch of the piston of the small cylinder was 3·627455 pounds, and that the pressure per square inch of the piston of the large cylinder was 1·258947 pounds, and that these pressures were constant for all speeds of piston. They are then the pressures which equilibrate the friction of the moving parts of the engine, and they must be supplied before the pistons can move, so that only what remain of the indicated pressures on the pistons of the two cylinders after the subtraction of respectively these friction pressures, are applied to the crank-pin and do work external to the engine. These remainders are termed the net pressures, and the horses-power calculated from them are termed the net horses-power.

The net horses-power developed by the engine, differs from the horses-power developed by the engine at the brake, by the friction of the load on the moving parts of the engine. When the load, or the factory shafting, is connected to the engine shaft, then a friction is thrown upon the journals and other moving parts of the engine, additional to the friction of the engine, *per se*, and proportional to the load, being a constant per centum of it. Consequently, the horses-power developed by the engine at the brake must be less to the extent of the horses-power required to overcome the friction of the load, than the net horses-power developed by the engine as above computed.

The total horses-power developed by the engine represents the entire work of all kinds performed by the steam, including, of course, the overcoming of the back pressure against the pistons of the two cylinders. This power is composed of the sum of the indicated horses-power developed in the two cylinders and the horses-power required to overcome the back pressure in them against their pistons. Now the pressures for the latter

horses-power are composed of the back pressures against the pistons, *exclusive* of the back pressures of the cushioned steam in the two cylinders. That is to say, the mean of the back pressures from the end of the indicator diagrams opposite the cushioning end up to the point in the stroke of the piston at which the cushioning commenced, is taken for the back pressure, and the length of the stroke of the piston through which the back pressure operates is the length between the same points. Consequently, the horses-power required to overcome the back pressure is calculated for a different stroke of piston than the stroke used for the calculation of the indicated and net horses-power, and different for each cylinder because the cushioning takes place at a different point of the stroke of the piston in each.

The horses-power required to cushion the steam, though expended in overcoming back pressure, is not included as such, because the cushioned steam thus obtained remains in the cylinder—not being exhausted to the condenser—and diminishes by that much the steam drawn from the boiler per stroke of piston. The economic value of this compressed steam, is equal to the compressing power expended upon it, therefore this power is restored during the succeeding stroke of piston by the cushioned steam. If the power expended in cushioning the steam was added to the power expended in overcoming the other back pressure, then the weight of steam so cushioned per stroke of piston, would have to be added to the weight drawn from the boiler per stroke of piston, in order to obtain a proper measure of the cost of the total horse-power. The cost of the indicated and net horse-power is, however, the number of units of heat in the weight of steam drawn from the boiler in a given time, whether there be cushioning or not, or to what extent.

For the small cylinder the calculation of the indicated, net, and total horses-power developed in it, is made for the entire area of its piston, the back pressure being computed down to the zero line. But for the large cylinders the calculations are so made for only the indicated and net horses-power. For the total power, the back pressure down to the zero line is computed as against only the ring or annular area which remains of the piston of the large cylinder after the piston of the small cylinder has been subtracted from it. The reason of this is that the back pressure, due to the piston of the small cylinder, has already been computed down to the zero line.

In the small cylinder, the total horses-power developed by the expanded steam alone, is computed for the mean pressure down to the zero line of the indicator diagram from the point in the stroke of the piston at which the cut-off closed to the end of the stroke, for the entire area of the piston, and for the length of stroke from the point in the stroke of the piston at which the cut-off closed to the end of the stroke.

For the large cylinder, the total horses-power developed by the expanded steam are the sum of the indicated horses-power developed in that cylinder, and of the horses-power expended in overcoming the back pressure, *including* that of the cushioning against the ring, or annular area, which remains of the piston of the large cylinder after subtraction of the piston of the small cylinder, and for the entire stroke of the piston, because all the steam which enters the large cylinder is used expansively from the beginning to the end of the stroke of its piston; and as the question is not horse-power *pro rata* to steam furnished from the boiler, but horses-power absolutely exerted by expanding steam, and for which there is *pro rata* liquefaction of steam in the cylinder, the power expended upon the cushioning must be included.

The cost of the indicated, of the net, of the total horse-power, and of the horse-power developed by the engine at the brake, is given in the table, both in pounds of feed water consumed per hour, and in Fahrenheit units of heat consumed per hour. The first is the measure popularly adopted, because easily expressed, and is sufficiently accurate for practical purposes, but the latter is absolutely correct under all circumstances. The difference in the two measures, for the conditions of ordinary practice, is not great, because the total heat of steam of different pressures within those limits does not vary largely, but the latter measure should be preferred when strict accuracy is needed, as in the case of comparisons. For the present experiments, however, in which the initial pressure upon the piston of the small cylinder, and the back pressure against the piston of the large cylinder, and the temperature of the feed water, varied to only a trifling degree, the difference in the two measures is nearly nothing.

In calculating the quantity of heat transmuted into the horses-power developed by the expanding steam (after the closing of the cut-off valve in the small cylinder), one Fahrenheit unit of heat is

taken as the thermal equivalent of 789¼ foot-pounds of work, which gives 41·811847 Fahrenheit units as the thermal equivalent of one horse-power (33,000 foot-pounds of work per minute); consequently that number multiplied by the number of horses-power developed by the expanding steam and by sixty, the number of minutes in an hour, gives the number of Fahrenheit units of heat transmuted per hour into this horses-power; the quotient of the division of this last product by the latent heat corresponding to the mean pressure of the expanding steam, gives the number of pounds of steam liquefied per hour in the cylinder to produce the horses-power developed there by the expanding steam.

As the experiments of the morning and afternoon of July 7th, and of the afternoon of July 8th, are repetitional, the mean of the three will be taken as expressing the economic results for the conditions under which they were made. The mean of the number of Fahrenheit units of heat imparted to the feed water in the boiler per hour during these experiments, is 1482830·027224. The mean of the number of Fahrenheit units of heat in the mixed condensing water and water of condensation withdrawn from the condenser per hour, is 1265923·565977. The difference is 0·146278 of the larger number.

For the above three experiments, the mean number of pounds of steam condensed per hour in the small and large cylinders, and in the receiver, to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut-off valve on the small cylinder, is 184·561707. The mean number of pounds of feed water pumped into the boiler per hour, is 1325·845679. The former number is 0·139203 of the latter number.

The difference between the above two determinations (0·146278 and 0·139203) is due to the loss of heat by radiation from the external surfaces of the jet condenser of the engine, and of the measuring tank; and to the fact that the water of condensation resulting from the liquefaction of the steam in the cylinders by the action of the interior surfaces of the latter, contains less heat when revaporized there than when originally vaporized in the boiler. The loss of heat by radiation from the external surfaces of the boiler, steam pipes and cylinders, must not be included in the

above comparison, because such loss is supplied as fast as it occurs by the fuel in the furnace. For the experiments in question, the loss of heat due to the above causes was (0·146278 — 0·139203 =) 0·007075 of the heat imparted to the feed water in the boiler.

For the experiment of the morning of July 8th, in which the steam was expanded 9·6444 times, instead of 6·2569 times as in the other experiments, the number of Fahrenheit units of heat imparted per hour to the feed water in the boiler was 1220557·870799; and the number present in the mixed condensing water and water of condensation withdrawn from the condenser per hour, was 1002735·052380. The difference is 0·178462 of the larger number.

The number of pounds of steam condensed per hour in the small and large cylinders, and in the receiver, to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut-off valve on the small cylinder, is 162·065479. The number of pounds of feed water pumped into the boiler per hour, is 1081·888922. The former number is 0·149799 of the latter number.

The difference between the above two determinations (0·178462 and 0·1497986) is due to the causes just explained as in the first case, but the loss of heat in this case amounts to (0·178462 — 0·149799 =) 0·028663 of the heat imparted to the feed water in the boiler.

That the loss of heat thus experienced should be proportionally greater in the latter case than in the former, is evident; first, because the loss of heat by radiation from the external surfaces of the condenser and measuring tank is constant for both cases, while the weight of feed water vaporized in the boiler was only eighty-two per centum in the second case of what it was in the first. Further, the cylinder condensations were much greater relatively to the feed water with the larger measure of expansion for the steam, than with the smaller one, and the revaporization of this water of condensation in the second case required less heat proportionally to weight than in the first case, owing to the greater expansion of the steam, and consequently smaller pressures in the cylinders.

The aggregate total horses-power developed by the engine represents the fuel consumed, for that power is the entire

mechanical effect produced by the fuel. Now, of this power, the aggregate net horses-power developed by the engine represents the commercially available part. The mean of the three experiments in which the steam was expanded 6·2569 times, shows that the net power was only 73 16175 per centum of the total power, the remaining 26·83825 per centum being expended in overcoming the back pressures against the pistons and the frictional resistances of the unloaded moving parts of the engine.

The frictional resistance cannot be lessened, but the back pressure can be materially. The back pressure against the piston of the large cylinder is easily understood; it is due to the pressure above zero in the condenser, increased by the resistance caused by the small area of the exhaust port, and by the tortuous passages connected with it. This back pressure can be lessened by larger area of port and by shorter and straighter passage, by a more thorough condensation of the steam, and by a better exclusion of air in leakage, but it will always be something, and any lessening due to more thorough steam condensation is certainly in part, and, under favorable circumstances, perhaps wholly offset by the correspondingly lessened temperature of the feed water. The larger exhaust port will have but little or no disadvantageous effect on the economy of the fuel, if the back pressure be properly cushioned; that is to say, cushioned up to the pressure in the valve chest.

But there is also a corresponding back pressure against the piston of the small cylinder, which is not so well understood, but which is just as injurious to the economy of the fuel as the back pressure against the piston of the large cylinder. This back pressure is the difference between the mean pressure in pounds per square inch above zero, shown by the bottom line of the indicator diagram taken from the small cylinder, exclusive of the cushioning, and the mean pressure in pounds per square inch above zero, shown by the portion of the top line of the indicator diagram taken from the large cylinder, lying between the commencement of the stroke of the piston and the point of cutting off the steam. This difference of pressure is the gap on the indicator diagram formed by properly combining the diagrams of both cylinders, and is caused by the too small areas of the exhaust port of the small cylinder and the steam port of the large cylinder. In the case of the experimental engine, these ports were much too small, and the

back pressure against the piston of the small cylinder was extremely and unnecessarily great.

The per centum of the aggregate total horses-power developed by the engine, transmitted at the brake to the factory shafting, was, as a mean of the three experiments in question, 69·07173, the difference between which and the per centum 73·16175 of the aggregate total horses-power realized as net horses-power is 4·09002 per centum, which is the proportion of the fuel absorbed by the friction of the load. This 4·09002 per centum, if referred to the net horses-power, becomes 5·59 per centum. The coefficient usually allowed is 7·50 per centum.

In the case of the experiment in which the steam was expanded 9·6444 times, the aggregate net horses-power were 71·76471 per centum of the aggregate total horses-power developed by the engine, and the horses-power transmitted at the brake were 69·12477 per centum, the difference 2·63994 is the proportion of the fuel absorbed by the friction of the load, and if referred to the net horses-power becomes 3·68 per centum.

The mean of the three experiments with the steam expanded 6·2569 times, gave the indicated horse-power for 17·103403 pounds of feed water consumed per hour, or for 19128·532189 Fahrenheit units of heat consumed per hour. The experiment with the steam expanded 9·6444 times, gave the indicated horse-power for 16·937819 pounds of feed water consumed per hour, or for 19108·790286 Fahrenheit units of heat consumed per hour. The results by the correct heat measurement are almost identical, and show that no economy of fuel followed increasing the measure of expansion with which the steam was used from 6·2569 to 9·6444 times.

This remarkable determination was obtained for experiments conducted with laboratory exactness by a number of highly educated experts of distinguished reputation, upon an excellent compound engine, and under the most favorable conditions for the large measure of expansion, namely: High initial pressure in the small cylinder, low back pressure in the large cylinder, cut off on both cylinders, and rapid reciprocating speed of piston, the cylinders being partially steam jacketed, and the steam being saturated. Of the truth of the result there can be no doubt, and it furnishes an upper limit for the economical expansion of steam.

When the comparison is made for the net horse-power, and for the total horse-power, the following are obtained: For the net horse-power there were consumed per hour 20717·401634 Fahrenheat units of heat as the mean of the three experiments in which the steam was expanded 6·2569 times; and 21100·310264 Fahrenheit units for the experiment in which the steam was expanded 9·6444 times. In this case, which is that of the commercially valuable horse-power, the smaller measure of expansion was 1·8 per centum more economical than the larger measure. For the total horse-power there were consumed 15157·303944 Fahrenheit units of heat as the mean of the three experiments in which the steam was expanded 6·2569 times; and 15142·576895 Fahrenheit units for the experiment in which the steam was expanded 9·6444 times; or almost exactly the same as with the lesser measure of expansion.

The calculated results from the experimental data show why the greater measure of expansion for the steam failed to produce any greater economy of the fuel than the lesser measure. The mean of the three experiments in which the steam was used with the lesser measure of expansion, shows that 24·777623 per centum of the steam entering the small cylinder remained condensed to water in it when the cut-off valve closed; and that in the experiment made with the larger measure of expansion, 43·813874 per centum of the steam which entered the small cylinder remained in it condensed to water when the cut-off valve closed.

The revaporization of the above water of condensation in the small cylinder under the lessening pressure due to the expanding steam, and at the expense of the heat in the metal of the cylinder and in the water of condensation, reduced the amount of condensation at the end of the stroke of the piston to 6·572403 per centum of the steam entering the small cylinder, for the smaller measure, of expansion; and to 11·674740 per centum for the larger measure. These percentages of water were revaporized in the small cylinder under the less pressure of the receiver when the exhaust valve opened, at the expense of the heat contained in the metal of the cylinder and contained in the water above the temperature due to the lessened pressure, and the steam thus produced passed into the receiver during the exhaust stroke of the piston, so that all the steam which entered the small cylinder finally passed into the larger one.

At the end of the stroke of the piston of the large cylinder, the water remaining in it due to the condensation of the steam, was 7·296654 per centum of the steam evaporated in the boiler for the smaller measure of expansion, and 10·521183 per centum for the larger measure of expansion, and these percentages of water were revaporized in the large cylinder under the less pressure of the condenser when the exhaust valve opened, and passed into the condenser during the exhaust stroke of the piston; the heat producing this revaporization being that which was contained in the metal of the cylinder and in the water of condensation. Of course, no work is obtained from the steam of the revaporized water of condensation that passes to the condenser during the exhaust stroke of the piston of the large cylinder, but the metal of the cylinder is chilled by this revaporization, so that it acts as a condenser to the next charge it receives from the small cylinder.

The re-evaporated steam which passed from the small cylinder to the large one during the exhaust stroke of the piston of the former, was utilized upon the piston of the latter, and by fitting the large cylinder with a lap cut-off steam valve, this re-evaporated steam was used expansively in that cylinder. The economic superiority of the compound engine over the simple one worked between the same boiler and condenser pressures, with the same measure of expansion and the same reciprocating speed of piston, is due to the fact that the steam condensed in the small cylinder by the interaction of its metal, is used upon the piston of the large one during its whole stroke, and expansively, too, if a cut-off be applied there.

The discovery that the cylinder of a steam engine acted alternately as a condenser and as a boiler, condensing a portion of the entering steam during its admission and revaporizing the resulting water of condensation during the period of expansion and during the exhaust stroke, which phenomena were caused wholly by the interaction of the metal of the cylinder, and would not have happened had the cylinder been constructed of a material which neither received nor imparted heat, was made a great many years ago by the writer, and stated and used in his published professional writings.* No one contested the discovery, or even used it, until

* Engineering Precedents. Vol. 2. 1859. Experimental Researches in Steam Engineering. 1861 and 1863.

quite lately; in fact, it was not for a long time believed. The discovery was made and proven in the only way possible, namely, by weighing the water fed into the boiler, and ascertaining, by means of the indicator, the weight of steam present as such in the cylinder at the point of cutting off and at the end of the stroke of the piston, adding to the last quantity the weight of steam condensed in the cylinder to furnish the heat transmuted into the power developed by the expanding steam, the latter obtained also by means of the indicator. The difference between the weight of water fed into the boiler and the weight of steam accounted for by the indicator at the above two points of the stroke of the piston, is the weight of steam condensed in the cylinder at those points by the interaction of the metal of which it is made. Of course, this quantity is to be slightly diminished by the weight of steam condensed in the steam pipe, valve chest and cylinder by external radiation. This, however, is unimportant in well-clad pipes and cylinders. If the water of condensation in the steam pipe flows back to the boiler instead of on into the cylinder, as it may do in whole or in part, depending on the inclination of the pipe and the velocity of the steam current, the allowance for the condensation by external radiation will be lessened that much.

No investigator other than the writer ever made experiments in this manner for a great many years after 1859, nor thought of ascertaining by measurement the difference of the weight of steam as given by the indicator and by the weight of water fed into the boiler, nor dreamed of attributing that difference to the interaction of the metal of the cylinder. The only cylinder condensation suspected was that which is obviously due to external radiation, and is a mere trifle under the conditions of good practice. This is the crucial point, and before any other experimenter can substantiate a claim to the discovery of cylinder condensation by the interaction of the metal of the cylinder, he must show the previous publication of an experiment in which that condensation is deduced from the comparison of the weight of steam accounted for by the indicator and the weight of water fed into the boiler. The writer is well acquainted with the literature of his profession, and is very confident that no such publication can be produced.

The writer made and published many such experiments in all their details on engines of different sizes and designs, deducing

from them, and for the first time, what may be called the *physiology* of the steam engine, as distinguished from an abstract mathematical conception of its theory, which had been exclusively in vogue before his work was given to the public, and still is largely accepted by professors; but his results, though now generally adopted by engineers, were for a great many years ignored, or rather silently disbelieved by all, because contrary to the then received ideas on the subject. Occasionally, some sorry attempts were made to ridicule them, but no one came forward to sustain the writer, much less to dispute his pretensions. The fact is, that for many years he stood alone. Not only was he not antedated in his discoveries, but no one seems for a long time after they were published with their full experimental proofs, to have understood them, or the important inferences deducible from them. Yet without the great underlying fact of cylinder condensation due to the interaction of the metal of the cylinder, there can be no correct theory of the steam engine; for without it there cannot be understood how the economy of the power, other things equal, is so greatly affected by mere size of cylinder as it is, becoming less as the size of the cylinder becomes less: how, other things equal, the economy of the power is affected by the reciprocating speed of the piston, becoming less as that speed becomes slower; how the gain theoretically due to greater measures of expansion for the steam, is not practically realized, the cylinder condensation increasing with increased measures of expansion, until the increased loss of steam by this condensation soon equals the gain due to the increased measure of expansion. Steam condensation in the cylinder by the interaction of its metal explains, and furnishes the only explanation of the economy to be derived from using superheated steam, from using steam jackets for the cylinders, and from compounding the cylinders.

The writer very early pointed out how quickly the limit of economy was reached by the expansion of saturated steam in a single cylinder, gave the reason therefor, supported it by abundant experimental proofs, and saw it rejected by every one for many years, and by many even now. He not only stated the facts qualitatively, but he experimentally gave the quantitative determinations for a great variety of cases. All that incredulity had to do was to repeat his experiments, but it was found much easier and

more agreeable to ignore them. In the few test repetitions that have been made, as in the present experiments, the writer's results were always reobtained, but the experimenters seemed to lack faith in their own work, and to think that some incomprehensible error had produced them.

No theory of the steam engine can have any value which is not based on the great fact of cylinder condensation by the interaction of the metal, and on the variations in the quantity of that condensation due to different conditions. Nothing else will explain the results of steam engineering; and here, as everywhere in industrial art, the properties of matter must be considered in connection with abstract laws, if practical results are to be correctly predicted, or even understood.

There is, perhaps, a necessity for stating that cylinder condensation by the interaction of the metal, is a totally different thing from the condensation in the cylinder of a portion of the steam to furnish the heat transmuted into the power developed by the expanding steam after the closing of the cut-off valve, because the late Professor Rankine, some years before his death, in a letter to the London *Engineer*, confused the two, and, knowing nothing of the former, seemed to think that by it the latter was meant.

In the following, Table No. 2, are given the data and results of the experiments made with the engine doing its regular work on two succeeding days, and continuing through the working hours of each, to test the accuracy of the results in Table No. 1, as it was supposed that a lengthened trial might give values different from those of a short one. The conditions for the experiments in both tables were sensibly the same, and so were the economic results.

Table No. 2, containing the data and results of the experiments made on the 22d and 23d of October, 1879, on a condensing compound engine, while doing its regular work in the factory of Mr. Antoine Herzog, at Colmar, in Alsace, Germany, by the Committee on Mechanics of the Industrial Society of Mulhouse, to ascertain the cost of the indicated horse-power in pounds of feed water consumed per hour:

	October 22, 1879.	October 23, 1879.
Duration of the experiments in consecutive hours and decimals of an hour,	12·3125	12·3403
Total number of double strokes made by the pistons of the engine,	65479·0000	65509·0000
Total number of pounds of feed water pumped into the boiler,	14095·0258	13922·4040
Steam pressure in the boiler in pounds per square inch above the atmosphere,	88·675	87·908
Number of double strokes made per minute by the pistons of the engine,	88·634856	88·475699
Number of pounds of feed water consumed per hour,	1144·773669	1128·206283
Pressure on the piston of the small cylinder at the commencement of its stroke in pounds per square inch above the atmosphere,	78·119	78·909
Mean indicated pressure on the piston of the small cylinder in pounds per square inch,	28·255	28·944
Mean indicated pressure on the piston of the large cylinder in pounds per square inch,	18·311	17·976
Mean back pressure against the piston of the large cylinder, exclusive of cushioning, in pounds per square inch above zero,	2·697	2·479
Indicated horses-power developed in the small cylinder,	23·113339	23·634452
Indicated horses-power developed in the large cylinder,	43·191937	42·325622
Aggregate indicated horses-power developed by the engine	66·305276	65·960074
Number of pounds of feed water consumed per hour per indicated horse-power developed by the engine,	17·265197	17·104382

HANGING OF CHARTS, DIAGRAMS, ETC., FOR DISPLAY, PRESERVATION, ETC.—In hanging charts, diagrams, maps, etc., for illustration of lectures, preservation, etc., a great desideratum is ease of removal, and substitution of others. Various devices, involving frames, pulleys, step-ladders, etc., have been employed, but the plan adopted by Professor Himes of Dickinson College, in the new Jacob Tome Scientific Building, seems to be peculiarly simple, effective, and cheap. As supports, wires are drawn around the room about six inches from the ceiling, or fourteen feet from the floor, fixed in eyelet screws, and supported at intervals as may be necessary, by means of screw hooks. Similar wires are drawn behind the lecture stand at intervals of several feet.

The diagrams are attached to strips of wood about a quarter-of-an-inch thick and an inch wide, by means of a spring clothes-clip at each end. One side of each of the clips is supplied with a hook made by a wire passed through it, and bent. An eyelet screw, of about one-half inch opening, is screwed into the middle of the strip, in front, through the chart, or the latter may be notched. By means of a wooden rod of suitable length, with a pin projecting about an inch from the end, thrust through the eyelet, the whole can be readily raised above the wire so as to bring the hooks down upon it. In a similar manner it can be taken down; the changing of the diagrams involving less trouble than by direct handling, and for simple preservation, several may be hung up on each other. The strip is readily removed, especially if the chart is notched, and the latter can be preserved in any other way. By a similar plan, suspensions from the ceiling for different purposes can be as readily made.

CYLINDER CONDENSATION IN STEAM ENGINES. AN EXPERIMENTAL INVESTIGATION.

BY CHAS. L. GATELY, M. E., AND ALVIN P. KLETZSCH, M. E.

[*Presented to the American Association for Advancement of Science; Ann Arbor Meeting, 1885, with an Introduction by* PROFESSOR R. H. THURSTON.]

INTRODUCTION.

In a paper read by the writer before the New York Academy of Sciences, several years ago,* the remark was made, referring to the nature of that function which determines the waste by "cylinder condensation:" "The function is evidently determined by the area of cylinder surface, time of exposure, variation of temperature of surfaces and quantity and character of the working steam."

The writer was compelled to assume the form of this function, and, after studying existing records of experiments, concluded that it was practically correct to take the amount of loss due to the cause above referred to as varying as the square root of the ratio of expansion. But, as remarked later,† "a comparison of the quantities of steam demanded to supply an engine thermodynamically 'perfect' with the actual quantities required by even the best of engines exhibits so wide a difference that it becomes obvious that the determination of the efficiency of an engine and the solution of the questions involving those of expenditure of heat are not problems in thermodynamics simply. The mathematical theory of the steam engine is not yet in so satisfactory a state—and cannot be until the correct theory of this transfer of waste heat can be introduced into it—that the engineer can often use it in every-day office work with much confidence, unless checked by direct experiment."

* "On the Behavior of Steam in the Steam Engine Cylinder, and on Curves of Efficiency;" JOURNAL OF THE FRANKLIN INSTITUTE, February, 1882.

† "On the Several Efficiencies of the Steam Engine, and on the Conditions of Maximum Economy." Trans. Am. Soc. of Mechanical Engineers, April, 1882; JOURNAL OF THE FRANKLIN INSTITUTE, May, 1882.

It had been the intention of the writer, for a long time previous to the publication of the paper referred to above, and of several other papers published at other dates, bearing upon the same general subject, to seek an opportunity to make a systematic investigation of this whole subject, and to determine, if possible, the laws governing the variation and the amount of cylinder condensation in steam engines, the speed of piston, the area of exposed surfaces, the ratio of expansion and the pressure of steam and its temperatures, varying within the range of familiar practice. It is not unusual to find steam engines in common use in which the speeds of piston vary from 200 up to 600 feet, or more, per minute, and 1,000 feet is becoming a not very unusual speed. Steam pressures range from twenty-five to 250 pounds per square inch in this country, and from two to ten atmospheres in Europe, while in exceptional cases, the pressures are vastly higher than the larger of the figures just given. The ratio of expansion varies from about two, in low pressure and the simpler kinds of engines, to ten, or even twenty, with high steam and superheating in machines designed with the intention of securing the utmost possible efficiency of fluid. In the latter, also, it is usual to find condensers in which a vacuum of twenty inches (two-thirds of one atmosphere) is maintained, in utter disregard of the fact that high temperatures of steam should be accompanied by a restricted amount of vacuum, and a warm condenser. The area of surface presented to the steam varies from about one square inch per pound of water per hour, in very small engines, to, perhaps, one quarter as much, in very large marine engines.

Before the date of the research to be described in the following paper, much had been done in this line of investigation, and some facts had been settled with a practically satisfactory degree of certainty. The work of Messrs. Clark, in England, and Hirn, in France, had shown the importance of this phenomenon of cylinder condensation, in the economy of the engine, and had given us some knowledge of the extent to which it affected the efficiency of certain classes of engine. The investigations of Isherwood had shown very clearly the extent, and the method of variation, of this waste, with varying ratios of expansion, in the marine engine of which that of the United States Steamship *Michigan* is the type. The studies of Escher had indicated the method of variation with

change of speed of piston; and other investigators had added something to the existing stock of knowledge. The writer, however, in common with many others in the profession, was desirous of securing an opportunity to investigate the matter more completely, and more systematically, than it had yet been possible, and sought to find a means of carrying out a plan which he had long before laid out. It was his desire to make an investigation upon an engine of fair size, of ordinary commercial construction, and under the usual conditions of every-day operation, first determining upon some one condition which should be made variable, all others being retained constant, then making this first condition constant, and another variable, and thus, one by one, determining the influences which, together, determine the total effect, measuring the extent of that effect, and learning the law of its variation. Thus it would be possible to ascertain the effect of variation of each of the determining conditions, as above, and to ascertain, and to express, the law of such variations, and the final resultant of all. To do this work properly required, evidently, great care, skill, patience, and familiarity with the steam engine and its management, and demanded a corresponding familiarity with the use of the apparatus of exact measurement to be applied in the work.

Such an opportunity finally offered, in the spring of 1884, when Messrs. Gately and Kletzsch, two young men who were attached to the Stevens Institute of Technology, found it possible to obtain an engine of 250 horse-power for purposes of experiment. An investigation, such as is above outlined, was at once planned, and was carried out during the later part of the spring and in the early summer of that year, and the results, as here given, were subsequently worked up as time and opportunity permitted, in such intervals of their business life as could now and then be secured by those engaged in the work.

Messrs. Gately and Kletzsch designed a "prony brake," which proved to be thoroughly well adapted to the very heavy work required of it, and which ran smoothly and successfully up to some 300 horse-power, a power which the writer had never before known to be applied to any brake controlling a steam engine. A weir was constructed, with which to measure the quantity of water of condensation, and the investigators were supplied with all needed

indicators, meters, thermometers and other apparatus from the Mechanical Laboratory of the Stevens Institute of Technology, then in charge of the writer, who had organized it for such purposes, as well as for commercial investigations, some years before. The work of investigation was performed mainly by the two gentlemen, who had assumed charge of this very important task. Fortunately, both had had experience in the use of instruments, such as were required in the work, both having been trained in the laboratories and the experimental department of the college, and Mr. Kletzsch having prepared himself with a view to taking charge of the mechanical laboratory of the Institute, a position which he has occupied up to the present time. They were aided by other trained assistants detailed from the college, and especially by Professor C. A. Carr, of the United States Naval Engineers, who, at that time, was on duty at Hoboken, as Professor of Marine Engineering and as Assistant Professor of Mathematics. The whole work was very carefully planned, the observations were made with great care and abundant skill, and the manner in which they have been worked up is evidence, equally, of the reliability of the observations, and of the ability with which the whole research was conducted and completed.

The results will be found of peculiar value and interest. In some cases, they confirm, in a very satisfactory manner, work previously done by the earlier investigators already referred to, and, in other instances, they give us entirely new and strikingly important deductions. The method taken of deriving these results, by a union of mathematical and graphical methods, will be seen to give great certainty, as well as to present the laws established in a manner which is most thoroughly satisfactory to the reader and to the investigator who may desire to make use of them in further research.

The whole report, as presented, finally, to the writer, is here given, as originally prepared, and is so complete in detail, and so well constructed, that it is unnecessary here to go into explanation, or to give a *résumé* of its conclusions. It will be observed that the method of variation of cylinder condensation, with variation of the ratio of expansion, is found to be precisely that indicated in earlier papers presented by the writer, and assumed in the theory of efficiency developed by him. The variation of condensation, with

changes of pressure and temperature, under usual conditions of practice, is found to be very moderate, and to follow a very simple law, so far as it can be traced. The waste with varying speed of engine is found, also, to be nearly as indicated by earlier writers; but the law is less exactly determined than in the case of varying expansion. Since, however, in all ordinary cases, in practice, the speed of engine and the boiler pressure are practically constant, in the regular operation of the engine, the most important part of the investigation is that relating to the ratio of expansion, which part of the work is most satisfactorily fruitful of result. The next most important matter is the determination of the variation of loss with varying speed of engine, and the results here reached are sufficiently exact to be very useful, both to the designer and the owner of engines; while the last investigation, that relating to variation with change of pressures, is very interesting as bearing upon the future of the now progressing advance in the direction of increasing pressures. The last two lines of research demand still further and corroboratory exploration. The results here reached must be regarded as applicable, in the theory of the steam engine, only provisionally, and as to be accepted finally, only after repeated experiment with other sizes and kinds of engine. This investigation, although most complete in plan and systematic in its prosecution, is but a single step in the work.

R. H. THURSTON

SIBLEY COLLEGE,
CORNELL UNIVERSITY, *Ithaca, N. Y., July 1, 1885.*

CHAPTER I.

OBJECT OF THIS RESEARCH.

The loss or waste, known as cylinder condensation, which occurs in the working of all steam engines, has been recognized by engineers as a more or less important factor, entering into the design of an engine ever since the time of Watt.

The solution of the problem of the method of variation of this condensation was suggested to the writers in the spring of 1884, by Professor R. H. Thurston, then of the Stevens Institute of Technology, Hoboken, N. J., as a most promising field for investigation in the department of steam engineering, not only from the fact of its vast importance to steam consumers at large, but also, because the best data existing on the subject were very incomplete and unsatisfactory.

Up to that time, although the general method of action was fairly well understood, the amount of loss which would occur from this cause with any given set of conditions, could not be predicted with that certainty which is essential in the designing of an engine. Thus we were led to the consideration of this question, and to attempt its experimental investigation. Our principal object was to furnish to the engineering profession a complete set of data, obtained from a good factory engine, running under conditions which were completely under our control, and such as to enable us to vary at pleasure the three most important factors entering into this question, viz., the ratio of expansion, the pressure and the piston speed.

Further than this, we desired to deduce the laws affecting this loss from a series of careful observations of the conditions governing each particular case of initial condensation.

Our limited time has rendered it necessary that several of the parts into which the subject might be subdivided should be left untouched, hence it is that we have given in considerable detail the construction of the special apparatus used, its arrangement when in use, a description of the engine, and the manner in which the trials were conducted, so that any one who may see fit to use the data here furnished, can satisfy himself as to its true value, with a better knowledge of all the details attending the trials, than he could get by simply an examination of the records of the observations.

I. CYLINDER CONDENSATION DEFINED—ITS METHOD OF ACTION.

As some difference of opinion still exists as to just how condensation takes place within the cylinder, and as to its causes, we may quote from several well-known authorities on the subject, in order that the different opinions of eminent engineers may be compared with results obtained by us, and which will be found summarized later.

Professor Rankine, in his "Steam Engine and other Prime Movers," pages 395–396, writes: "That liquefaction does not, when it first takes place, directly constitute a waste of heat or of energy; for it is accompanied by a corresponding performance of work. It does, however, afterwards, by an indirect process, diminish the efficiency of the engine; for the water which becomes liquid in the cylinder, probably in the form of mist and spray, acts as a distributor of heat, and equalizer of temperature, abstracting heat from the hot and dense steam during its admission into the cylinder, and communicating that heat to the cool and rarefied steam which is on the point of being discharged, and thus lowering the initial pressure and increasing the final pressure of the steam, but lowering the initial pressure much more than the final pressure is increased; and so producing a loss of energy, which cannot be estimated theoretically."

Professor Jas. H. Cotterill, in his treatise on "The Steam Engine Considered as a Heat Engine," page 246, says:

"When the volume of steam actually delivered from a steam cylinder at release is compared with the volume of dry steam at the terminal pressure, corresponding to the amount of feed water used, it is always, or nearly always, found that the first is far less than the second, showing that at the end of the stroke the steam discharged from the cylinder must contain more or less water, which is either re-evaporated during exhaust, or is carried out with the exhaust steam in the shape of suspended moisture.

"Some of this is no doubt due to the fact that the steam supplied by the boiler is rarely dry, but in general the difference in question is far too great to be thus accounted for, and it is, therefore, necessary to suppose that liquefaction takes place after the steam enters the cylinder. Moreover, when the expansion curve drawn by an indicator is examined, it is almost always found, even when

the greatest care has been taken to eliminate the disturbing causes, to show that evaporation takes place during expansion.

"Now these unquestionable facts can only be explained by supposing that liquefaction takes place during the admission of the steam to the cylinder, and evaporation during expansion and exhaust.

"This alternate liquefaction and evaporation is chiefly due to the action of the sides of the cylinder, in many cases combined with the effect of water remaining in the cylinder after exhaust is completed.

"It is in the first place clear that the amount of steam liquefied on admission must depend on the area of the surface exposed to the steam. It is true that other circumstances may, or rather must, have influence, and especially the time during which the contact of the steam with the surface lasts, and the temperature of the surface; but the first consideration is the actual area of the surface."

Referring to Chief Engineer Isherwood's report of the Marine Engine trial, made by a board of Naval Engineers, Professor Cotterill continues:

"We may draw the following conclusions from them:

"(1.) The initial condensation, and the heat absorbed per square foot of admission surface, increase with the ratio of expansion.

"(2.) The initial condensation at high rates of expansion may exceed fifty per cent."

Professor Thurston writes:*

"Of all the heat furnished to the engine by the boiler, a certain part, definite in amount and easily calculated, when the power developed is known, is expended by transformation into mechanical energy, another part, equally definite, and easily calculated, also, is expended as the necessarily occurring waste which must take place in all such transformations, at ordinary temperatures of reception and rejection of heat; still another portion is lost by conduction and radiation to surrounding bodies; and finally, a part, often very large in comparison with even the first and most important of these quantities, is wasted by transfer within the

* "On the Several Efficiencies of the Steam Engine;" Trans. Am. Soc. of Mechanical Engineers; JOURNAL OF THE FRANKLIN INSTITUTE, 1882.

engine from the induction to the eduction side, or from 'steam to exhaust' without any useful effect being derived therefrom. This last transformation is what is properly known as cylinder condensation.

"The amount of this loss increases with wet steam, and is diminished by any expedient, as steam jacketing, or super-heating, which prevents the introduction or production of moisture in the midst of the mass of steam in the cylinder.

"It is generally assumed, in the usual theory of the engine, that the expansion of the working fluid takes place in a cylinder having walls impermeable to heat, and in which no losses by conduction or radiation can occur; but it is impossible, in practice, so far as is known, to secure a working cylinder whose material will fulfil these conditions. The consequence is, that since the steam or other working fluid enters at a high temperature, and is discharged at a comparatively low temperature, the surfaces of the cylinder, cylinder heads and piston are one moment charged with heat of a high temperature, and at the next instant exposed to lower temperatures, are drained of their surplus heat, which heat is then rejected from the cylinder through the exhaust and wasted. In the steam engine, the loss by the method here referred to is rarely less than one-fourth, in unjacketed cylinders, and is often more than equal to the whole quantity of heat transformed into mechanical energy.

"As the range of temperature worked through in the engine increases as the quantity of steam worked per stroke diminishes, and as the time allowed for transfer of heat to and from the sides and ends of the cylinder and piston is increased, the magnitude of this loss increases. Hence the use of high steam of a high ratio of expansion, and of low piston speed, tend to increase the amount of this waste."

2. INFLUENCE OF CYLINDER CONDENSATION UPON EFFICIENCY—METHOD OF SECURING ECONOMY—INFLUENCE ON THE RATIO OF EXPANSION AT MAXIMUM EFFICIENCY.

By reference to Table 1, line 31, it will be seen that, of all the steam that passed into the cylinder from the boilers, from twenty-two to fifty per cent. absolutely passed through the engine to the exhaust without producing any useful dynamical effect upon the

piston. From this it can readily be understood at what point the engine of the present day fails, and why so small a percentage of the energy stored in the fuel consumed by the boilers is realized in giving motion to the machinery of our mills. Any saving effected in this direction would be felt throughout the world in the cheapening of the manufactured products, and he who accomplishes it will be considered as great a public benefactor as he who first conceived the idea of utilizing the steam itself.

In considering the theory of the steam engine thermodynamically, expansive working of the steam is found to produce economy, but the effect produced by cylinder condensation is not considered, hence it is that the extent to which expansion should in theory be carried is found to be greatly in excess of that found to give the best results in practice. This can readily be understood by referring to *Plate I*, when it will be seen that, as the ratio of expansion increases, or as the fraction of the stroke completed up to cut-off diminishes, the percentage of condensation rapidly increases, the angle made by the curve representing the variation with the axis of X, on which the percentage of condensation is laid off, becoming less and less as the cut-off is earlier. Therefore, a point must exist at which, expansion being carried beyond it, the loss due to condensation will exceed the gain of work done by the expanding fluid.

Professor Thurston, in a paper on the "Several Efficiencies of the Steam Engine," read before the American Society of Mechanical Engineers, Philadelphia, 1882,* writes: "The efficiency of engine has been often studied by authorities considered as standard, but almost invariably as a problem in thermodynamics, simply; and the losses of heat occurring in consequence of the working of steam in a cylinder composed of a good conductor of heat have been left unnoted, although frequently the most important of all the expenditures of heat taking place in the engine.

"In a non-conducting cylinder, the maximum efficiency of fluid would be secured if the ratio of expansion were made nearly the quotient of initial by back pressure; while the efficiency of the en-

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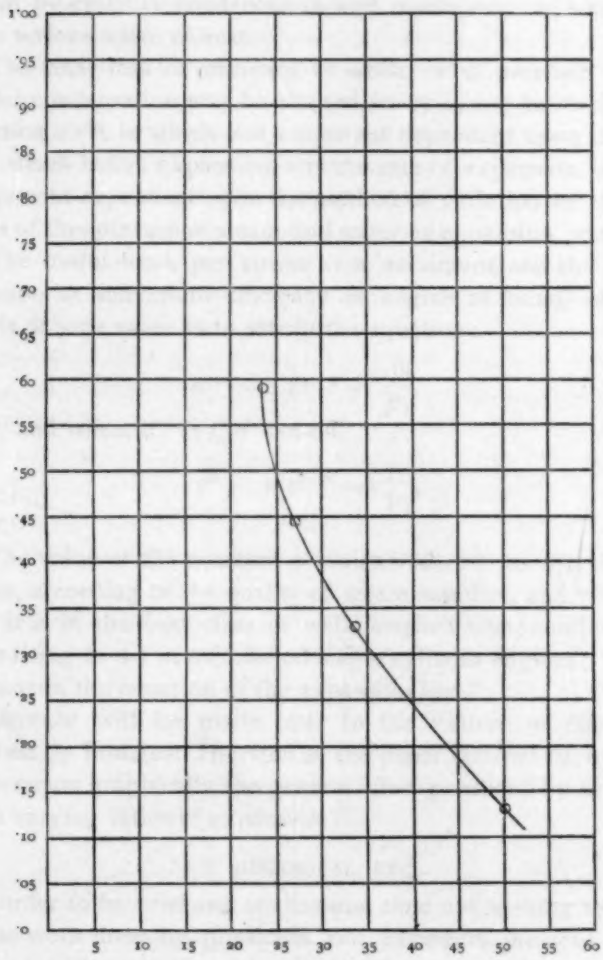
*Trans. Am. Soc. of Mechanical Engineers, 1884; JOURNAL OF THE FRANKLIN INSTITUTE, 1882.

PLATE I.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gine would be made a maximum when the ratio is made nearly equal to the quotient of initial pressure by the sum of all useless resistances. When, however, as is always the case in practice, the steam is worked in a metallic cylinder, the best ratio of expansion is made very much smaller, and the efficiency of the engine is greatly reduced by cylinder condensation and re-evaporation, which produce a serious waste of heat.

"The total loss of efficiency of work, or of pressure, due to cylinder condensation may be allowed for by taking for its value the expansion $a\, r^{m}$, in which a is a constant dependent upon the state of the steam before expansion, r is the rate of expansion, and ^{m} is an exponent dependent upon the method of variation of the properties of the mixture of steam and water as expansion progresses.

"The useful work per stroke is a maximum, and the ratio of expansion at maximum efficiency of engine is found, when the latter is of such value as to satisfy the equation:

$$r^{-n} - a\, r^{m-n} = \frac{p_0}{p_1},$$

nearly, and when, if c is the 'cut-off,'

$$c^{n} - a\, c^{n-m} = \frac{p_0}{p_1},$$

nearly.

"The value of the constant a varies from 0·1 to 0·2, in good engines, according to the quality of steam supplied, and ^{m} may be taken at o in the best class of well-designed compound engines, and as rising to 0·5 in unjacketed single cylinder engines; ^{n} is the exponent in the equation of the expansion line."

Reference will be made later to the "curve of efficiency," described by Professor Thurston in the paper referred to, by which he represents graphically the general effect produced by condensation at varying ratios of expansion.

3. HISTORICAL, ETC.

In order to be brief and at the same time not wishing to underrate the work done by physicists and engineers previous to our time, we will quote, from Dr. R. H. Thurston's "Theory of the Steam Engine,"* the following historical outline:

* Report of British Association for Advancement of Science, Montreal Meeting, 1884; printed in extenso, p. 569.

"The limitations of the thermodynamic theory of the heat engines, and of its application in the design and operation of such engines, were first discovered by James Watt a hundred years ago and more. They were systematically and experimentally investigated by Isherwood, in 1855 to 1865, were observed and correctly interpreted by Clark in 1855 and earlier, and were revealed again by the experiments of Hirn, and by those of Emery and many other recent investigators on both sides of the Atlantic. These limitations are due to the fact that losses occur in the operation of steam engines which are not taken into account by the hitherto accepted theory of the engine, and have no place in the thermodynamic treatment of the case.

"James Watt discovered this cause of the limitation of the efficiency of the steam engine. He not only discovered the fact of the existence of this method of waste, but experimentally determined its amount in the first engine ever placed in his hands. It was in 1763, that he was called upon to repair the little model of the Newcomen engine, then and still in the cabinets of the University of Glasgow. Making a new boiler, he set up the machine and began his experiments. He found, to his surprise, that the little steam cylinder demanded four times its own volume at every stroke, thus wasting, as he says, three-fourths of the steam applied, and requiring four times as much 'injection water' as should suffice to condense a cylinder full of steam. It was in the course of this investigation that he discovered the existence of so-called 'latent heat.' All of Watt's first inventions were directed toward the reduction of this immense waste. He proposed to himself the problem of keeping the cylinder 'as hot as the steam that entered it;' he solved this problem by the invention of the separate condenser and the steam jacket, and thus the discovery of the limitation of the thermodynamic theory here noted was the source of Watt's fame and fortune.

"John Smeaton, a distinguished contemporary of Watt, and perhaps the most distinguished engineer of his time, seems to have been not only well aware of this defect of the steam engine, but was possibly even in advance of Watt in attempting to remedy it. He built a large number of Newcomen engines between 1765 and 1780, in some, if not many of which, he attempted to check loss by this now familiar 'cylinder condensation' in engines, some

of which were five and six feet in diameter of cylinder, by lining pistons and heads with wood. This practice may not be practicable with the temperatures now usual; but no attempt has been made, so far as is known to the writer, to follow Smeaton in his thoroughly philosophical plan of improvement. Cylinder condensation remains to-day, as in the time of Smeaton and Watt, the chief source of waste in all well-designed and well-constructed heat engines.

"It is a curious fact, and one of great interest as illustrating the gulf formerly separating the philosopher studying the steam engine and working out its theory from the practitioner engaged in its construction and operation in the earlier days of engineering, that, notwithstanding the fact that this waste was familiar to all intelligent engineers, from the time of the invention of the modern steam engine, and was recorded in all treatises on engine construction and management, the writers on the theory of the machine have apparently never been aware that it gives rise to the production, in the working cylinder, of a large amount of water mingled with the steam. In fact, it has often been assumed by engineers themselves that this water is always due to 'priming' at the boiler. Even Rankine, writing in 1849–50, while correctly describing the phenomenon of cylinder condensation, made the mistake of attributing the presence of the water in steam cylinders to the fact of condensation of dry steam doing work by expansion, apparently not having noted the fact, that this would only account for a very insignificant proportion of the moisture actually present in the average steam engine. He considers incomplete expansion the principal source of loss, as do usually other writers on thermodynamics.

"Thomas Tredgold, writing in 1827, who, but little later than Carnot, puts the limit to economical expansion at the point subsequently indicated and more fully demonstrated by De Pambour, exaggerates the losses due to practical conditions, but evidently does perceive their nature and general effect. He also shows that under the conditions assumed, the losses may be reduced to a minimum, so far as being dependent upon the form of the cylinder, by making the stroke twice the diameter.

"The limit of efficiency in heat engines, as has been seen, is thermodynamically determined by the limit of complete expansion. So well is this understood, and so generally is this assumed to

represent the practical limit, by writers unfamiliar with the operation of the steam engine, that every treatise on the subject is largely devoted to the examination of the amount of loss due to what is always known as 'incomplete expansion'—expansion terminating at a pressure higher than the back pressure in the cylinder. The causes of the practical limitation of the ratio of expansion to a very much lower value than those which maximum efficiency of fluid would seem to demand, have not been usually considered either with care or with intelligence by writers thoroughly familiar with the dynamical treatment, apart from the modifying conditions here under consideration.

"Watt, and probably his contemporaries and successors, for many years supposed that the irregularity of motion due to the variable pressure occurring with high expansion was the limiting condition, and does not at first seem to have realized that the cylinder condensation, discovered by him, had any economical bearing upon the ratio of expansion at maximum efficiency. It undoubtedly is the fact that this irregularity was the first limiting condition with the large, cumbrous, long-stroked, and slow-moving engines of his time. Every accepted authority from that day to the present has assumed, tacitly, that this method of waste has no influence upon the value of that ratio, if we except one or two writers, who were practitioners rather than scientific authorities.

"Mr. D. K. Clark, publishing his *Railway Machinery*, in 1855, was the first to discuss this subject with knowledge, and with a clear understanding of the effects of condensation in the cylinder of a steam engine upon its maximum efficiency. Cornish engines, from the beginning, had been restricted in their ratio of expansion to about one-fourth as a maximum, Watt himself adopting a 'cut-off' at from one-half to two-thirds. Hornblower, with his compound engine competing with the single cylinder engines of Watt, had struck upon this rock, and had been beaten in economy by the latter, although using much greater ratios of expansion; but Clark, a half century and more later, was the first to perceive precisely where the obstacle lay, and to state explicitly that the fact that increasing expansion leads to increasing losses by cylinder condensation, the losses increasing in a much higher ratio than the gain, is the practical obstruction in our progress toward greater economy.

"Clark, after a long and arduous series of trials of locomotive engines, and prolonged experiment looking to the measurement of the magnitude of the waste produced as above described, concludes: 'The magnitude of the loss is so great as to defeat all such attempts at economy of fuel and steam by expansive working, and it affords a sufficient explanation of the fact, in engineering practice, that expansive working has been found to be expensive working, and that, in many cases, an absolutely greater quantity of fuel has been consumed in extended expansion working, while less power has been developed.' He states that high speed reduces the effect of this cause of loss, and indicates other methods of checking it. He states that 'the less the period of admission relative to the whole stroke, the greater the quantity of free water existing in the cylinder.' His experiments, revealing these facts, were, in some cases, made prior to 1852. But the men handling the engines had observed this effect even before Clark; he states that they rarely voluntarily adopted 'a suppression of above thirty per cent.,' as they found the loss by condensation greater than the gain by expansion. Describing the method of this loss, this author goes on to say that 'to prevent entirely the condensation of steam worked expansively, the cylinder must not only be simply protected by the non-conductor—it must be maintained by independent external means, at the initial temperature of the steam.' He thus reiterates the principle expressed by Watt three-quarters of a century before, and applies it to the newly-stated case.

"The same author, writing in 1877, says: 'The only obstacle to the working of steam advantageously to a high degree of expansion in one cylinder, in general practice, is the condensation to which it is subjected, when it is admitted into the cylinder at the beginning of the stroke, by the less hot air surfaces of the cylinder and piston, the proportion of which is increased so that the economy of steam by expansive working ceases to increase when the period of admission is reduced down to a certain fraction of the stroke, and that, on the contrary, the efficiency of the steam is diminished as the period of admission is reduced below that fraction.' The magnitude of this influence may be understood from the fact that the distinguished engineer, Loftus Perkins, using steam of 300 pounds pressure, and attaining the highest economy known up to his time, found his engine to consume 1·62 pounds of

fuel per hour per horse-power; while this figure is now reached by engines using steam at one third that pressure and expanding about the same amount, and sometimes less.

"Mr. Humphreys, writing a little later than Clark, shows the consumption of fuel to increase seriously as the ratio of expansion is increased beyond the very low figure which constituted the limit in marine engines of his time.

"Mr. B. F. Isherwood, a chief engineer in the United States Navy, and later Chief of the Bureau of Steam Engineering, seems to have been the first to attempt to determine, by systematic experiment, the law of variation of the amount of cylinder condensation with variation of the ratio of expansion, in unjacketed cylinders. Experimenting on board the United States Steamship *Michigan*, he found that the consumption of fuel and of steam was greater when the expansion was carried beyond about one-half stroke than when restricted to lower ratios. He determined the quantity of steam used, and the amount condensed, at expansions ranging from full stroke to a cut-off at one-tenth. His results permit the determination of the method of variation, with practically satisfactory accuracy, for the engine upon which the investigation was made, and for others of its class. It was the first of a number of such investigations made by the same hand, and these to-day constitute the principal part of our data in this direction. The writer, studying these results, found that the cylinder condensation varied sensibly as the square root of the ratio of expansion, and this is apparently true for other forms and proportions of engine. The amount of such condensation usually lies between one-tenth and one-fifth the square root of that ratio, if estimated as a fraction of the quantity of steam demanded by a similar engine having a non-conducting cylinder.

"The state of the prevalent opinion on this subject, at the time of this work of Clark and of Isherwood, is well expressed by the distinguished German engineer, Dr. Albans, who, writing about 1840, says of the choice of best ratio of expansion: 'Practical considerations form the best guide, and these are often left entirely out of view by mathematicians. Many theoretical calculations have been made to determine the point, but they appear contradictory and unsatisfactory.' Renwick, in 1848, makes the ratio of initial divided by back pressure the proper ratio of expansion, but

correctly describes the effect of the steam jacket, and suggests that it may have peculiar value in expansive working, and that the steam may receive heat from a cylinder thus kept at the temperature of the 'prime' steam. John Bourne, the earliest of now acknowledged authorities on the management and construction of the steam engine, pointed out, at a very early date, the fact of a restricted economic expansion. Rankine recognized no such restriction as is here under consideration, considered the ratio of expansion at maximum efficiency to be the same as that stated by Carnot and by other early writers, and only perceived its limitation by commercial considerations, a method of limitation of great importance, but often of less practical effect than is the waste by condensation. In his *Life of Elder* (1871), however, he indicates the existence of a limit in practice, and places the figure at that previously given by Isherwood, for unjacketed engines. By this latter date, the subject had become so familiar to engineers that a writer in *London Engineering* in 1874, contemns writers who had neglected to observe this limitation of efficiency as indulging in 'mediæval twaddle.'

"A few writers on thermodynamics finally came to understand the fact that such a limitation of applied theory existed. M. G. A. Hirn, who, better than probably any authority of his time or earlier, combined a knowledge of the scientific principles involved with practical experience and experimental knowledge, in his treatise on thermodynamics (1876), concludes: '*Qu'il est absolument impossible d'édifier* a priori *une théorie de la machine à vapeur d'eau douce d'un charactère scientifique et exact*,' in consequence of the operation of the causes here detailed. While working up his experiments upon the performance of engines, comparing the volume of steam used with that of the cylinder, he had always found a great excess, and had, at first, attributed it to the leakage of steam past the piston; but a suggestion of M. Leloutre set him upon the right track, and he came to the same conclusion as had Watt so many years before. He explains that errors of thirty, or even up to seventy per cent. may arise from the neglect of the consideration of this loss. Combes had perceived the importance of this matter, and De Freminville suggested the now familiar expedient of compression, on the return stroke as nearly as possible to boiler pressure, as a good way to correct the evil. The

matter is now well understood by contemporary writers, and it has become fully agreed, among theoretical writers as well as among practitioners, that the benefit of extended expansion in real engines can only be approximated to that predicted by the theory of the ideal engine, by special arrangement having for their object the reduction of cylinder waste, such as superheating, 'steam jacketing' and 'compounding.'

"Professor Cotterill has given more attention to this subject than any writer up to the present time. He devotes a considerable amount of space to the study of the method of absorption and surrender of heat by the metal surfaces enclosing the steam, constructs diagrams which beautifully illustrate this action, and solves the problems studied by him with equal precision and elegance of method. He summarizes the experimental work done to the date of writing, and very fully and clearly exhibits the mode of transfer of heat past the piston without transformation into work. Professor Cotterill's treatise on the steam, 'considered as a heat engine,' is invaluable to the engineer.

"Thus the theory of the steam engine stands to-day incomplete, but on the verge of completion, needing only a little well-directed experimental work to supply the doubtful elements. Even these are becoming determined. Isherwood gives facts showing waste to be proportional, very nearly, if not exactly, to the square root of the ratio of expansion, and Escher, of Zurich, has shown the loss to be also nearly proportional to the square root of the time of exposure, or, in other words, to the reciprocal of the square root of the speed of rotation, and it only remains to determine the method of variation of loss with variation of range of temperature to give the whole of the necessary material for the construction of a working theory which will enable the engineer to estimate, in advance of construction, the economic performance of his machine. There will undoubtedly be much more to be done in constructing an exact theory involving all the physical changes occurring in the working of the heat engines familiar to us; but it will yet be done, and probably very soon. It is the hope of the writer that experiments made under his direction recently may furnish the needed data, as the result of the first systematic research directed to that end; but if this should prove not to be the fact, it cannot be long before direct investigation will secure all essential knowledge.

When this is the case, the remarks of those distinguished physicists and engineers Hallauer and his great teacher Hirn will be no longer well based upon apparent fact.

"Says Hirn, in his memorable discussion with Zeuner, in regard to this subject: '*Ma conviction reste aujourd'hui ce qu'elle était il y a vingt ans, une théorie proprement dite de la machine à vapeur est impossible; la théorie expérimentale, établie sur le moteur lui-même et dans toutes les formes où il a été essayé en mécanique appliquée peut seule conduire à des résultats rigoureux.*'

"Chronologically considered, it is seen that the history of the growth of the theory of the steam engine divides itself distinctly into three periods, the first extending up to the middle of the present century, and mainly distinguished by the attempts of Carnot and of Clapeyron to formulate a physical theory of the thermodynamics of the machine; the second beginning with the date of the work of Rankine and Clausius, who constructed a correct thermodynamic theory; and the third beginning a generation later, and marked by the introduction, into the general theory, of the physics of the conduction and transfer of that heat which play no part in the useful transformation of energy. The first period may be said to include, also, the inauguration of experimental investigation, and the discovery of the nature and extent of avoidable wastes, and attempts at their amelioration by James Watt and by John Smeaton. The second period is marked by the attempt, on the part of a number of engineers, to determine the method and magnitude of these wastes by more thorough and systematic investigation, and the exact enunciation of the law governing the necessary rejection of heat, as revealed by the science of thermodynamics. The third period is opening with promise of a complete, and practically applicable, investigation of all the methods of loss of energy in the engine, and of the determination, by both theoretical and experimental research, of all the data needed for the construction of a working theory.

"The writer would therefore make the classification of these successive stages in the progress here described, thus:—

"(1.) Primary period.—That of incomplete investigation, and of earliest systematic, but inaccurate, theory.

"(2.) Secondary period.—That of the establishment of a correct thermodynamic theory, the *theory of the ideal engine.*

"(3.) Tertiary period.—That of the production of the complete theory of the engine, of the true *theory of the real engine.*"

4. PROPER METHOD OF INVESTIGATION—CONDITIONS DEFINED.

The deficiency of complete data on the subject in hand has already been alluded to. The most complete are those embodied in the report of Chief Engineer B. F. Isherwood, U. S. Navy, of the trials made by a board of Naval Engineers, on Marine Engines.*

These trials were made particularly to ascertain the effect of varying ratio of expansion upon cylinder condensation; for this reason and from the type of the engines on which they were made, the data are very incomplete and unsatisfactory.

To determine by experiment the part which each of the factors, ratio of expansion, pressure and piston speed plays in causing the great loss of heat found to occur in the engine cylinder, due to condensation, by noting the effect produced by variation in all of them at the same time, or in other words, to make a series of trials, beginning with a known value for ratio of expansion, pressure and piston speed within the power of the engine and boilers to maintain and keep them constant throughout one trial of any desired length, and then for the next trial give different values to each of these three quantities, and likewise for each succeeding trial, and, from the data obtained, deduce the law governing each case, would be extremely difficult, if not absolutely impossible.

After much thought on the plan to pursue, the writers decided that determining the effect of each factor separately, and afterwards, if possible, combining the results obtained into one expression representing the effect which would be produced by giving different values to one or all of the three variables which would of necessity enter into the equation, would without doubt be the simplest and best plan and at the same time bring success within a probability.

* Engineering Researches.

CHAPTER II.

5. MACHINERY AND APPARATUS.

The engine used in our experiments is located at the Upper Rubber Mill, of the New York Belting and Packing Company, Sandy Hook, Conn.

It was built by Wm. A. Harris, Providence, R. I, in 1880, and is of the type known as the Harris-Corliss engine.

It is 42-inch stroke of piston, 18-inch diameter of cylinder, and is fitted with the ordinary jet condenser and reciprocating air-pump, 9-inch diameter of cylinder by 10-inch stroke; but, as will be seen from the logs of trials, the condenser can be disconnected and the engine worked without condensation.

The injection water was obtained from the head-race, leading from the mill dam to the water-wheel, and entered the condenser on an average temperature of about 68° Fahrenheit. The valve gear consists of a wrist-plate, operated by the eccentric, and to which are connected four rods, operating all the valves, the two steam valves being closed by vacuum dash pots.

The steam valves are tripped at the proper moment by small cams on the stem of each valve, which are operated by the governor, thus securing the automatic cut-off, which is a recognized essential.

The exhaust of engine is closely connected to the condenser by a seven-inch pipe, and steam is conveyed from the boilers by a five-inch pipe, well protected with hair felt and canvas. The cylinder was not jacketed, but was covered with a non-conducting substance and wooden staves. The engine is furnished with a pulley fly-wheel, 14 feet in diameter and 25-inch face, belted to a jack-shaft, which in turn is coupled to the main shaft of the mill; but in these trials this coupling was removed and the work supplied by a brake, which will be described later on. When in actual use, the engine is run about four months out of the twelve, water-power being sufficient for the remainder, its work consisting in giving motion to grinders and sheeters, ordinarily used in the preparation of rubber prior to its vulcanization into the various forms in which it is commercially used. It was found upon a preliminary examination to be in excellent condition, steam and exhaust valves were tight and, in fact, the seats showed no wear whatever.

Leakage of the piston was particularly looked for, since in these trials particularly, its existence would be fatal; but it was found to be absolutely tight.

6. THE BOILERS.

Steam was furnished in most of the trials by two boilers, but in two instances it was found necessary to use a third. They were built at the Bridgeport Boiler Works, Bridgeport, Conn., in 1880, and are a standard type of the horizontal fire tube boiler, the only notable peculiarity about them being that the gases from the furnace are prevented from directly circulating toward the back and by a bridge wall built of fire brick; but instead are carried upwards through two large holes in the forward end of the crown sheet, to the combustion chamber, which really corresponds to the furnace of an upright boiler.

There they are drawn through the tubes, deflected downwards underneath the back part of the boiler, and then up the chimney, passing through a feed-water heater in their escape.

The principal dimensions are as follows:

Diameter of shells,	42 inches.
Length,	17 feet 6 "
Number of tubes, each boiler,	37
Diameter of tubes,	3 "
Length of tubes,	14 feet.
Heating surface, each shell,	192·5 square feet.
Heating surface of tubes, each boiler,	407·0 " "
Heating surface of heads, each boiler,	19·2 " "
Total heating surface,	618·7 " "

7. SPECIAL APPARATUS.

Brake.—As we believe this is the largest brake of its kind ever constructed, the design is given in detail and described separately.

Indicators.—The indicators used were "Thompson," two in number, Nos. 549 and 340, manufactured by the American Steam Gauge Company, and, tested by them after the trials were completed, the springs were found to be correct, the pistons tight, and the indicators to be in good order. The motion was obtained indirectly from the cross-head of the engine, through a strip of wood oscillating about a pin and fastened in the ceiling of the engine room and linked to a three-quarter-inch iron rod fastened rigidly

to the cross head. The line was fastened to the stick at a point sufficiently far from the centre of oscillation to give the required length of card, and which length remained sensibly constant during the trials.

In running under variable pressures, the spring was changed for each trial, so as to give a sufficient height of card to readily admit of measurement; but when the pressures were constant through a whole set the same spring was used.

The Weir—The writers thought it advisable, as a means of check, to ascertain the number of heat units carried off by the condensing water. To compute this, we must first know the weight of water discharged from the condenser, and, secondly, its rise in temperature. The most correct method for getting this necessary data, was by the use of a tumbling bay or weir, through which all the condensing water is made to pass, a method as yet in very little use in this country, but more familiar in Great Britain.

The appliances required are very simple, and can be readily fitted to any engine. It consisted of a strong water tight wooden box, 5 feet long, 2 feet 6 inches deep, and 2 feet wide.

It is fitted near one end with a series of perforated transverse partitions, while at the other end an aperture 8 x 10 inches is cut out, forming a notch, there being fixed outside of the notch a thin brass plate $\frac{1}{8}$ inch thick, having an opening in it corresponding with that in the box, but somewhat smaller (6 x 8 inches) and having its edge bevelled outwards, so that the water flowing through encounters very little resistance from friction. One foot and a half back from this notch, and across the top of the box, is fastened a strip of wood $2\frac{1}{2}$ x 3 inches, to which is bolted a micrometer screw, made with the greatest nicety and capable of being read to the $\frac{1}{10\,000}$th part of an inch, on the end of this is fastened a hook gauge, such as is commonly used in measuring the flow of water, its *o* being on a level with the horizontal edge of the notch.

The whole apparatus was placed in a level position under the end of the discharge pipe from the condenser, so that the issuing water would have a clear fall of 14 inches. The water entering the box passed through, over and under the transverse partitions, being thus thoroughly mixed in its course, and at the same time

sending it quietly and smoothly under the hook gauge, where its head above the bottom of the notch was measured.

The Meter.—A two-inch Worthington Meter was introduced in the feed pipe leading from the feed water heater, so that the weight of water passing into the boilers could be easily calculated, the height in the glass gauges was noted at the beginning of the trial, and at the end it was made to exactly agree, so that the meter readings in cubic feet reduced to pounds would exactly represent, after deducting that which passed through the calorimeter, the amount of water evaporated by the boilers and sent over to the engine, in the form of steam. Between the boilers and the meter, and close to the latter was fitted a drip pipe, furnished with a stop-cock, so that the temperature of the feed water could be read at pleasure.

The Calorimeter.—A one-inch pipe lead from the steam dome of one of the boilers (and indirectly from the other, as both boilers were of course connected) to a calorimeter. This was obtained from the Stevens Institute of Technology, and is the one used there in all their boiler trials. It consists of a wooden tank, lined inside with zinc and packed with hair felt to prevent radiation, the tank being capable of holding 100 pounds of water. Inside is a coil of ½-inch pipe connecting with the pipe leading from the boiler by a three-way cock, so that the steam can be passed through it, or into the air when desired.

At each test, a known weight of water is introduced into the tank surrounding the coil and being thoroughly mixed by means of a float which can be moved through the liquid. By this simple apparatus, the percentage of water passing over with the steam to the engine is easily determined, and the method of calculating it from the data obtained from observation will be found in Chapter IV.

8. METHOD OF STANDARDIZING THE SPECIAL APPARATUS AND RESULTS.

The vacuum and steam gauges used in the trials were manufactured by the American Steam Gauge Company, and the latter was carefully standardized by us before and after the trials by comparison with one of Shaw's Mercurial Pressure Gauge, and found to exactly correspond with it at all pressures between the limits at which we worked, these being the only ones at which we subjected it to the comparison.

(*To be Continued.*)

THREE NEW PORTRAITS OF WASHINGTON.*

COMPOSITE PHOTOGRAPHY.

The FRANKLIN INSTITUTE has so recently done me the honor to invite an exhibition and description of composite photography, and the proceedings on the occasion were so fully reported in the August number of the JOURNAL, that no extended description of the plate here published may now be required. These Washington portraits (marked "*Composite*" on the plate), made by combining the various representations that have come down to us, ought to have a peculiar historical value, in that they are, each, the sifting of the testimony of a number of eye-witnesses. In the case of the upper composite—the profile—seven artists, whose names are attached to the surrounding originals, present, as a body, their impressions of the great man's appearance; and, as each artist has only one-seventh of a showing in the result, no unsupported individual notion can possibly assert itself. The same may be said of the two other composites at the bottom of the plate, except that each of these combines five originals.

The most complete demonstration of the value of this method of photography lies in the fact that, while the conceptions of the individual artists are so diverse, the combined testimony of the several groups is to one effect; for the composite of the seven and each composite of five look like one man, which cannot be said of the individual heads.

So much admiration has been expressed about the beauty of the composite on the lower right of the plate that we are having it finished finely, of life size, in crayon, for exhibition at the "NOVELTIES." Care will be taken not to disturb the peculiar sketchy look of the composite, and to change none of the suggestions of its multifarious origin, nor any characteristics which, by their prominence indicate a weight of authority for their retention. If this purpose be carried out with the skill we hope for, the result ought to be a Washington with higher credentials than any that has preceded it, for it will have the authority of fourteen contemporaneous

*See Frontispiece.

artists, adjusted to agreement. This portrait or one founded on a similar method will, in time, be esteemed as the truest likeness of the Father of his country.

Every reader, convinced of the reasonableness of what is here said, will recognize that composite photography has before it an important future, in those departments of portraiture to which it is adapted. And it is pleasant to reflect that our FRANKLIN INSTITUTE has been so prompt to illustrate its claims before the public.

W. CURTIS TAYLOR.

THE MICROSCOPIC STRUCTURE OF IRON AND STEEL,*

BY F. LYNWOOD GARRISON, Philadelphia, Pa.

[*Read at the Chattanooga meeting of the American Institute of Mining Engineers, May, 1885.*]

It is not intended to make in the present paper any deduction or to formulate any theories from the results obtained by experiments. The further expenditure of considerable time and labor would be required to obtain a sufficient basis for positive assertions as to the microscopic structure of the different varieties of iron and steel, and the structural changes which take place in them. It is therefore my purpose at this time to offer simply a synopsis of the general results arrived at in a few months' work, a brief description of apparatus used, and a few hints as to the preparation and preservation of the material which it is intended to investigate.

The study of the microscopic structure of iron and steel is not altogether new. Some attention has been given it in both England and Germany. But the foreign publications on the subject have thus far been confined, so far as I am aware, to two papers, one by Herr Martens, of Berlin, contributed to the *Verein zur Befoerderung des Gewerbfleisses*, and the other a lecture by Dr. H. C. Sorby, of Sheffield. Dr. Sorby, it seems, was induced to investigate the subject as bearing on the structure of meteoric iron; and the results he obtained are certainly very interesting. At the Boston meeting of this Institute, in February, 1883, Mr. J. C. Bayles called attention to the subject in a neat and exceedingly

*From Advance Sheets of the Transactions of the American Institute of Mining Engineers.

interesting paper, in which the work of Messrs. Martens and Sorby was summarized,† and original suggestions were added.

It is at present difficult to say what will be eventually the practical value of the microscope, thus employed, in the sciences of engineering. The *rôle* which it seems most likely to play is that of an adjunct to the testing-machine, and not (as some have supposed) a rival to the chemical laboratory. That it will be a most valuable accessory seems, to say the least, highly probable.

I need hardly go at length into the details of preparing the material for examination. Mr. Bayles has described the process in such a plain and comprehensive manner, that if his instructions are carefully followed, one need not encounter any serious obstacles after a little experience and the expenditure of a considerable amount of time and patience. Patience and cleanliness are the two most important attributes to be acquired by a student, if he desires success in work of this character. A deficiency in either will be sure to spoil his work, and in the end he will give it up in disgust wondering what has been the cause of his failures.

In grinding the specimens, it is quite unnecessary that they should be ground to an extreme thinness and mounted in Canada balsam as microscopical objects are usually preserved. This entails a vast amount of labor, to no end whatever. A good and accurate photograph, once obtained, is usually sufficient for any reference that might be desired in the future; besides, with a little care the etched surfaces of the objects can be preserved from rust by simply rubbing a few drops of kerosene oil over them with a soft chamois-skin and then placing them in a tightly-corked phial.

The size of the objects to be examined under the microscope, may vary considerably; but the sizes found most convenient range from $\frac{1}{4}$ down to about $\frac{1}{16}$ of an inch in thickness and from an inch to $\frac{1}{6}$ of an inch in sectional area. If the specimens are extremely thin, there is often much difficulty in mounting them properly on a slide, and getting the etched surface perfectly parallel to the object-glass. After the surface has been sufficiently treated with acid, and shows under the microscope no further traces of scratches made in the grinding, it should be carefully dried and cemented to a glass slide with wax or cement, great care being taken to have

†Transactions. vol. xi., p. 261.

it in the proper plane parallel to the object-glass: otherwise, it will be impossible to make a satisfactory photograph.

The greatest difficulty encountered in pursuing the study of the structure of materials is that of making accurate and satisfactory records of what is seen under the microscope. To effect this, the only accurate and quick means is to photograph. Hence the student must not only be a good microscopist, but also understand the theory and practice of photography, an accomplishment which every engineer will find it useful to acquire. The subject of photo-microscopy, although a comparatively new one, is somewhat extensive, and it would not be within the scope of this paper to enter upon it in detail. A few hints, however, may be of value. The camera should not be large; the most convenient size will be found to be one using 4¼ x 5½-inch plates, and having a bellows capable of being extended several feet. Instantaneous dry plates only should be used, for experience has shown that no others will serve the purpose half as well; and they should be developed with a weak solution of an alkaline or a ferrous oxalate developer. It is found that when an instantaneous plate is used, it is better to make a comparatively long exposure and use a weak developer than to make a short exposure and use a strong developer.* The time of exposure depends upon many conditions, such as the clearness of the atmosphere, the intensity with which the etched surface reflects light, the quality of the objectives, and the sensitiveness of the gelatine plate. The above conditions are frequently so complex that it would be impossible to say just how long the exposure should be made; the student must learn to judge this by experience. A safe limit, however, may be placed at not over twenty seconds, when good *direct* sunlight can be obtained and a perfectly instantaneous plate used.

The proper illumination of the object to be photographed under the microscope is somewhat difficult and requires considerable practice to effect it. It must also be borne in mind that the best results can be obtained only by direct sunlight, unobscured by clouds or mists. When the object to be photographed has been carefully adjusted on the stage of the microscope, and the focuss-

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*A developer diluted to *one-half* its usual strength will be found to give excellent results.

ing has been approximately made, the table (holding both the camera and microscope) is placed near the window, so that the sunlight may strike *directly* upon the object. The fine focussing is then made by means of a cord passing around in a groove in the periphery of the fine adjustment-screw of the microscope. The image on the ground glass should be perfectly clear and well-defined in all its details. To effect this, a focussing glass, such as is ordinarily used by photographers, may be used with advantage in adjusting the finer details. The use of a condensing lens depends upon the ability of the etched surface to reflect light. Thus, for instance, hard steel reflects light so well that a condensing lens is necessary, while in the case of pig, cast, or wrought iron, its use is absolutely essential. For further particulars on this subject, Dr. Sternberg's work, *Photo-Micrographs and How to Make Them*, is recommended.

In selecting a microscope, though the question of cost is of primary importance to many, it should be remembered that a good instrument once obtained will, with proper care, last a life-time. Such an instrument, under ordinary circumstances, should not cost less than $35 or $40, although a good second-hand one may be obtained for less. The most important points to be observed in selecting a microscope, with which it is intended to take photo-micrographs, are as follows:

(1.) The stand should be of the best workmanship and material; there should be no "shake" or lateral motion; in the adjustment of the focus, there should be no "lost motion"—that is, the focus should instantly change with the slightest motion of the milled heads—and for photography there *must* be a universal joint, for inclination.

(2.) The instrument must be provided with a fine adjustment-screw with a groove turned in its periphery.

(3.) The mirror under the stage should be so constructed that it can be made to swing over, around, and above the stage, thus affording a means of more intense illumination to the object than otherwise could be obtained with opaque substances. The objectives should be of the very best quality. Experience has shown that it is poor economy to use any others. Beck's ⅔-inch and 1½-inch will be found to give excellent results, and a range of powers quite sufficient for any ordinary work. There are a number of

makers of objectives, all about equally good, but as I have used Beck's lenses only, I say nothing about the rest. Two great points to be observed in selecting an objective are, that it has a good "*penetration*" and an extra lens (to be used only when photographing), so that the visual and chemical foci may be made to coincide. The correction of Beck's is effected in the 1⅔-inch by substituting for the back-stop a double convex lens of 8-inch focus. and for the ⅔-inch a double convex lens of 5-inch focus. Higher powers need no correction, and lower ones than 1½ inch are rarely employed.

The quality of the eye-piece is of secondary importance, as it is never used in photographing. The attachment of a mechanical stage to the microscope is a great convenience and economizer of both time and patience, although it is not an absolute necessity. Dr. Carpenter's work and Dr. Phin's *Hints on the Selection and Use of the Microscope*, together with Dr. Sternberg's work are recommended to those who may desire further details on the several subjects.

Of all the varieties of iron and steel, pig and cast iron are the most difficult to prepare for examination.

Fig. 1 represents a No. 3 pig iron (gray) as it appears under a power of fifty diameters. The specimen was prepared and etched with the greatest possible care, so that it may be safely taken as a good example of pig iron, ranging from No. 1 to No. 4, when magnified to that degree. It will be seen to consist of a heterogeneous mixture of metallic iron and long, narrow, black plates of graphite. It does not appear to bear the slightest trace of any crystalline structure. The straight black lines which seem to stand out in relief are the graphite plates. Owing to the comparatively high power used in this case, the slightest inequalities of the etched surface cause an unevenness in the focussing; hence the obscurity of some parts of the plate. By close observation of some parts, however, the structure and the graphite plates can be made out. In many cases of pig and cast iron the graphite plates seem to group themselves in bunches or lumps. This seems to be more characteristic of cast than of crude pig iron. It is not unlikely that the second melting and slow cooling enables the graphitic carbon to separate itself more readily in that way.

White pig iron exhibits a highly crystalline structure, as will

FIG. 1.

No. 3 Gray Pig Iron. × 50 Diameters.

FIG. 2.

White Pig Iron. × 20 Diameters.

FIG. 3.

Longitudinal Section of Rolled Bar Iron. × 45 Diameters.

FIG. 4.

Cross-Section of Rolled Bar Iron. × 45 Diameters.

FIG. 5.

Longitudinal Section of a Bolt of Clapp-Griffiths Steel. × 45 Diameters.

FIG. 6.

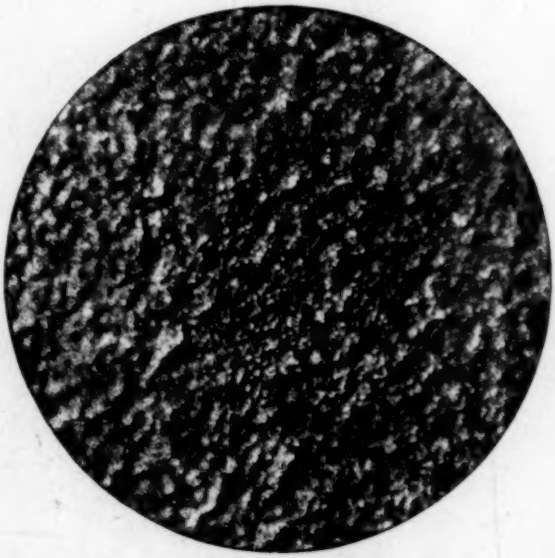
Cross-Section of a Bolt of Clapp-Griffiths Steel. × 45 Diameters.

(*Garrison.*)

FIG. 7.

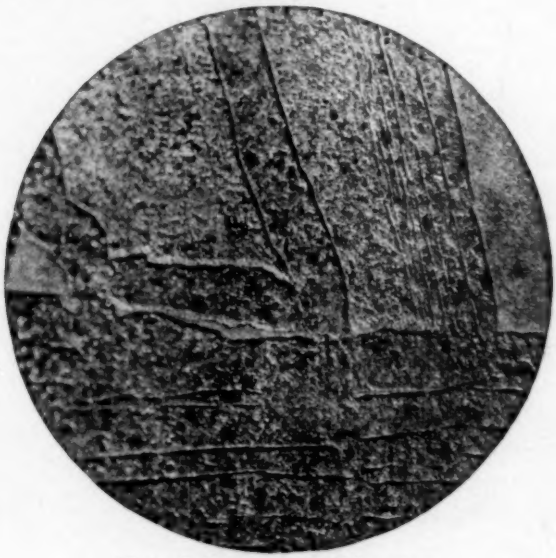
Crucible Tool Steel. × 45 Diameters.

FIG. 8.



File Steel. × 40 Diameters.

FIG. 9.



Meteoric Iron. × 20 Diameters.

FIG. 10.

"Burnt Out" Grate Bar of Cast Iron. (*A*) The Cast Iron Unaltered by the action of the Fire. (*B*) The Cast Iron Changed into Hard *Steel* by the action of the Fire. × 45 Diameters.

be seen in *Fig. 2*. The intensity of the crystallization depends very much upon the degree of chilling. Thus, in a large casting made in a metallic mould, the outer surface, which comes in contact with the mould, will be found to exhibit a high crystalline structure (such as shown in *Fig. 2*), while the inner part, which has cooled slowly, will show very little, or, perhaps, no crystalline structure. This highly crystallized white iron exhibits, even under a high power, only a comparatively small number of graphitic plates. The crystalline structure in some cases is irregular, while in others the crystals are regularly arranged, with their long axes normal to the surface of the mould. The plates of graphite will be found to be arranged parallel to the lines of crystallization.

Wrought iron or mild steel exhibits a fibrous structure, running in the direction in which it has been rolled. *Fig. 3* shows the structure of a fine quality of rolled bar iron. The fibre is distinct, and shows numerous furrows and cavities, due to working, and the presence of intermingled slag during the rolling. Wrought iron (not steel) does not show, even under a power of 100 diameters, the slightest trace of crystalline structure. It has been held by Percy and other authorities that the fibres were simply drawn out crystals. I have tried, in numerous instances, to determine if such really was the case, and, although I have examined many longitudinal and cross-sections of various grades, I cannot find that there exists any foundation for such a view.

Fig. 4 shows a cross-section of the same material as *Fig. 3*. The furrows of *Fig. 3* will be seen to be replaced by irregular cavities in the cross-section.

Figs. 5 and *6* show, respectively, a longitudinal and a cross-section of a bolt made of Clapp–Griffiths steel. The difference in structure between iron and steel can be readily seen in this case, the latter showing a fibrous and yet finely granular structure, characteristic of steel.

Hard, or tool steel, presents a structure entirely different from any of the preceding. It is highly crystalline, uniform in structure, and shows no lines of weakness, or any tendency in the crystals to develop themselves in any given direction. *Fig. 7* is a high-grade crucible tool steel magnified forty-five diameters.

Fig. 8 is the usual grade of file steel magnified to the same

degree. The latter differs from crucible tool steel only in being somewhat more compact and harder. All steels exhibit a similar characteristic structure, which enables a person, with practice, to judge of their relative qualities by a simple comparison of their compactness, lustre and crystalline structure.

Fig. 9 shows the structure of meteoric iron, which is quite different from any of the artificial irons, or alloys of iron and nickel. The peculiar lines so prominent in the figures are characteristic of meteoric iron, and are commonly known as "Widmannstatten lines." I have not been able to detect these lines in all varieties of meteoric iron. It seems that if the iron be very impure, and contain but a small amount of nickel, there is little or no tendency to develop them.

One of the most interesting and peculiar changes of iron into steel, which have come under my notice in connection with this subject, is exhibited in *Fig. 10*. It shows a section made from a "burnt-out" grate bar of ordinary cast iron. The left hand side (*A*) shows the cast iron unaltered by the action of the heat; the right side shows where the cast iron has been completely changed to hard steel, which resists the file quite as much as any tool steel. On examination of the part of the bar which came most in contact with the fire, I found that the entire surface was changed to hard, compact steel, with a thickness of about one-tenth of an inch. The most remarkable point, to my mind, is that the line of demarcation is so sharply defined, thus showing little or no intermediate stage of decarbonization. The altered part (*B*) shows a structure decidedly characteristic of hard tool steel (compare with *Fig. 8*). In the unalterable part (*A*), the structure of cast iron is quite apparent, the groups and clusters of graphite plates being readily distinguishable (compare with *Fig. 1*). As the reasons for such a remarkable change of structure might cause considerable speculation and much difference of opinion, it would, perhaps, be better to defer its discussion for another opportunity and a separate paper.

In conclusion, it is most earnestly desired that this paper may excite a friendly criticism and discussion of the subject, by which we may learn the opinions and experiences which others have had of the physical properties of iron and steel. From any persons interested, I should be very glad to receive such specimens or information as they may see fit to send.

BOOK NOTICES.

CHEMICAL PROBLEMS. By Dr. Karl Stammer. Translated from Second German Edition, by W. S. Hoskinson, A. M. P. Blakeston, Son & Co. 1885.

This little work, of about a hundred pages, is published without a preface. It is, however, evidently intended to be a guide to the teacher, rather than to the student, since all the problems have their respective answers given in the end of the book, and, moreover, there is very little explanatory comment in any part of the work, such as a learner in the science would require.

The problems are essentially stoichiometrical questions, and relate to all of the commonly occurring elements—each element having several pages of questions, which are varied in every conceivable manner.

Indeed, the book, so far as it goes, is quite exhaustive, and is a good example of German thoroughness in compilation.

There are two objections to the work; a large portion of the questions are exactly such as arise in the practice of analytical chemistry, and have to be solved daily in both gravimetric and volumetric work. It is quite certain that a much deeper impression would be made upon the student who attacks such questions one by one as they occur in laboratory work, than would be made by attempting to figure through Dr. Stammer's several thousand "problems." Such an effort would prove a weariness to the flesh, and would probably exhaust all enthusiasm the student might feel for the science.

Secondly, there is rather a superabundance of material offered. A few pages, for instance, would be quite sufficient to thoroughly explain the metric system; yet, all through the work, answers are required in grams, pounds, kilograms, etc. This is a waste alike of the printer's ink and of the reader's patience.

A few errors are noticeable; chromite is spoken of (page 74) as though it were a pure spinel type, containing only one atom of ferrous and of chromic oxides, with no other bases present. Such a chromite does not exist. Again, the atomic weight of aluminum is stated to be 27·4; Mallet has shown that its correct weight is 27. The book is clearly printed, and appears to be well translated. Problem 42, page 71, being an exception in this latter respect.

H. P., JR.

THEORIE ELASTISCHER KÖRPER. Eine Einleitung zur Mathematischen Physik and Technischen Mechanik. Von Dr. Jacob J. Weyrauch. Leipzig. 1884. And

AUFGABEN ZUR THEORIE ELASTISCHER KÖRPER. Von Dr. Jacob J. Weyrauch, Leipzig. Druck und Verlag von B. G. Teubner. 1885.

The want of a comprehensive theory, which should include all phenomena due to the elastic properties of solid, fluid and gaseous bodies, has induced the author to develop a general theory, and he has accomplished this task as only a man can do who fully abandons himself to it, only striving to fathom the subject to the bottom, no matter whether the benefit of his labors will accrue to himself or to others.

He found it necessary to introduce new concepts and new functions. After first developing the most general laws, he deduces from them the laws bearing on special cases and special phenomena. The deductions are throughout analytical, and since the nature of the subject demanded the triaxial system, the formulæ appear mostly in triplets.

All bodies are supposed to consist of material elements which exert upon each other pressures, that as a rule, are not at right angles to the contact surfaces. These forces are resolved in right-angular tensions and parallel shifting forces, and, by examining their tendencies in their entirety, the respective mathematical formulæ are deduced. As these forces produce motion, the theory of vibration is established.

In the *Aufgaben* the application is shown of the fundamental formulæ, to the solution of general problems bearing on the subject of elastic bodies.

H. B.

DAS PRINCIP VON DER ERHALTUNG DER ENERGIE SEIT ROBERT MAYER. Zur Orientirung. Von Dr. Jacob Weyrauch. Leipzig, Druck und Verlag von B. G. Teubner. 1885.

This is a concise treatise on the conservation of the energy of forces with special reference to the time and authorship of the succeeding discoveries, and the general progress of this science, and especially to the literature on the subject. It is therefore a desirable hand-book for those who not only wish to become fully acquainted with the details, but also with the history of this principle which, according to Herbert Spencer, is one of the three great discoveries of this century.

H. B.

THE PHILADELPHIA INSURANCE CHART, for 1885, has been issued by J. H. C. Whiting. It contains a large amount of condensed information, being not only a perfect directory as to insurance companies, agents and brokers, but all the places of interest in the city are given, with information respecting them, the names of streets, their length, width, etc. "How Philadelphia Burns" is shown in a table, evidently made with much care and labor, giving the number of all the vast manufacturing establishments of the city, value of such property attacked by fire, the losses, insurance and ratios, not only for 1884, but for the period of 1874–1884, inclusive. This has now become a standard insurance annual, and will be found also valuable to the statistician and historian. To be obtained from the Review Publishing and Printing Company, Northwest corner Fourth and Walnut Streets. N.

THE FIREMAN'S GUIDE. A Hand-Book on the Care of Boilers. By Technologföreningen T. I. Stockholm. Translated from the Third Edition and Revised by Karl P. Dahlstrom, M. E. 12 mo. Published by E. & F. N. Spon. London and New York.

This practical book of only twenty-eight pages, including preface, besides being of most respectable technical origin, is conspicuous by extreme conciseness and clearness in statements and recommendations. Such brevity is valuable here, because engineers and firemen will not read extensive treatises.

The chapters comprehend firing and economy of fuel, feeding, low water, pressure, cleaning and blowing out; also, general directions for repairs, prevention of accident, etc. A summary of rules at the conclusion is plain and easy to be remembered. This compact hand-book of boiler management should be owned by every manufacturer or user of steam boilers; and such persons would advance self-interest by presenting a copy thereof to their engineers and firemen. N.

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